HEATING AND VENTILATING

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AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION

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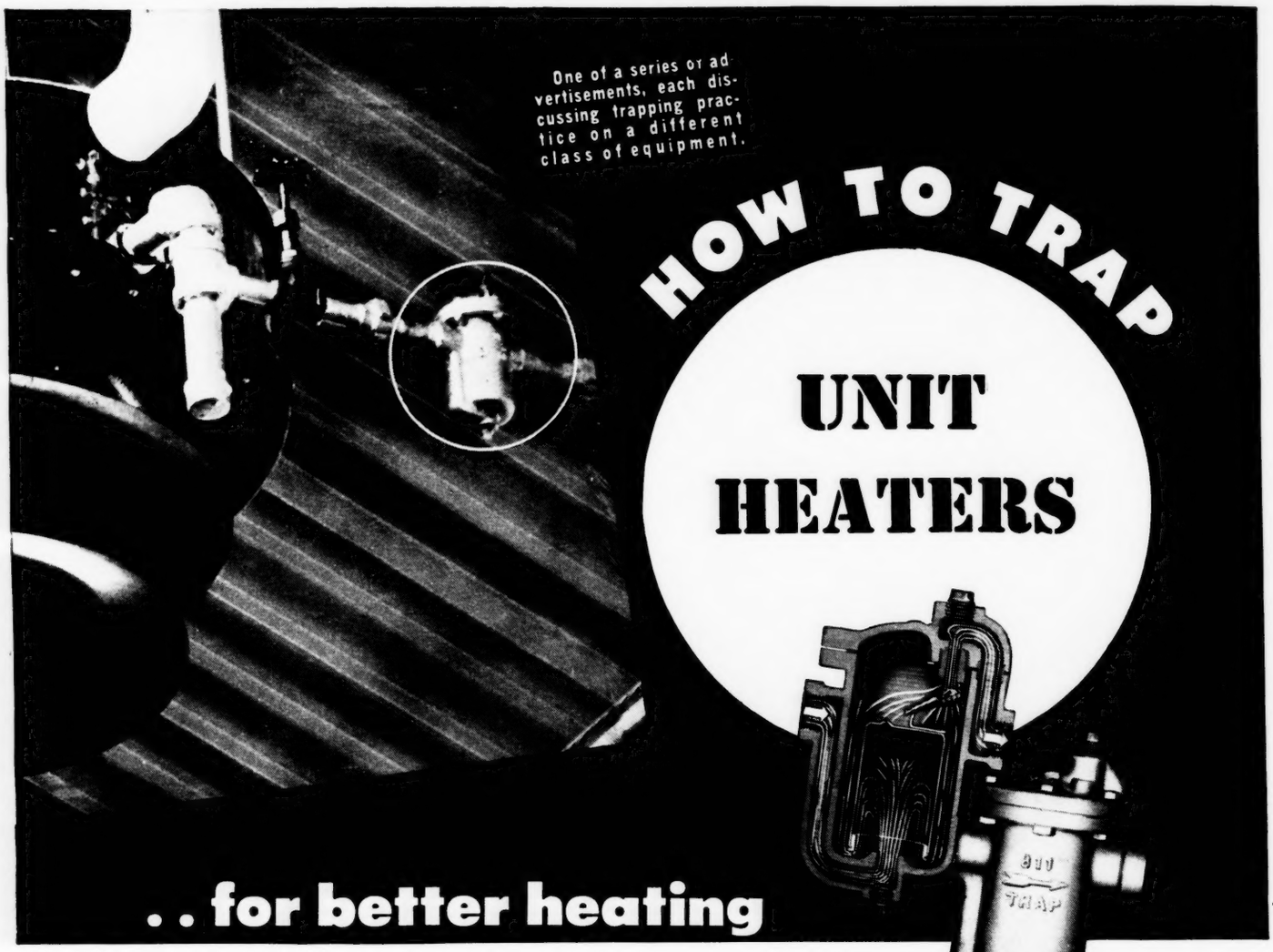
Reducing Condensate Temperatures . page 90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VENTILATING

A MONTHLY MAGAZINE FOR ENGINEERS AND CONTRACTORS CONCERNED WITH COMMERCIAL, INDUSTRIAL AND INSTITUTIONAL AIR CONDITIONING, REFRIGERATION, AIR SANITATION, PIPING, HEATING AND VENTILATION

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THIS MONTH'S COVER

Under-window convector blankets window exposure with warm air to offset radiant heat loss to cold glass surface. Photo courtesy Modine Mfg. Co. The case for convector heating comfort is presented by A. G. Dixon beginning page 57.

WHAT WE MAKE

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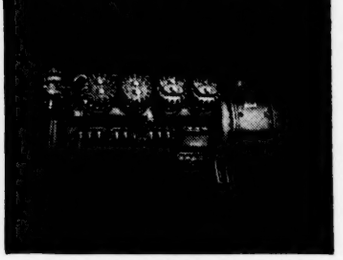
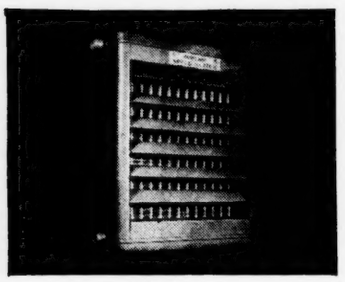
The **SILENTVANE FAN** is one of the most versatile of all air-moving units—performing thousands of jobs in heating, ventilating, and mechanical-draft applications. Available in 19 sizes, this is just one of the many types of blower and exhaust fans listed.

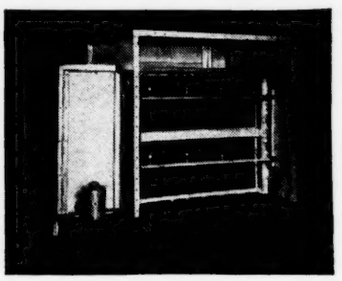
AIR-CONDITIONING



HERMETICALLY-SEALED COMPRESSOR—heart of the air-conditioning system—is designed to save power and prevent refrigerant losses, with completely sealed-in construction and direct motor drive. No shaft seals, belts, pulleys or couplings.



The **SPEED HEATER**—a compact, efficient unit providing fast, economical heat—is typical of Sturtevant's complete line of space heaters for commercial and industrial applications. Whatever your needs, you'll find a Sturtevant unit for the job.



The **SURFACE DEHUMIDIFIER**—another vital component—maintains just the right amount of moisture for comfort or industrial air conditioning. These precision-engineered units are available in 47 standard sizes, to meet every conditioning requirement.

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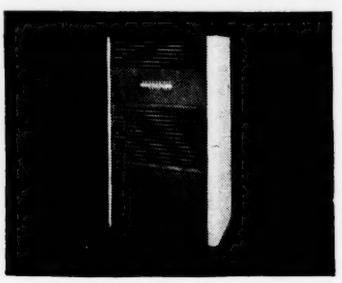

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Washington News

LORING F. OVERMAN

Coal Strike
G. O. P. May Act
Vets' Housing Stalled
Production Climbs

Of longest-range interest to the heating, ventilating and air-conditioning industry is the possibility that the latest John L. Lewis strike move may cause industrial, domestic and commercial consumers to give a second thought to fuels other than coal.

For the second time in six months, coal users have been subjected to the uncertainty of supply; the economic losses of actual shortages, and the personal sense of frustration that goes with being at the mercy of a labor leader who doesn't shrink from a test of the comparative powers of himself and his Government. To say that fuel users do not like to be placed in such a position every few months is an understatement.

New Selling Point

Segments of the H & V industry concerned with designing, constructing and merchandising oil and gas fired equipment may have John L. to thank for a new selling point—the dependability of fuel supply—but the chances are that the immediate price they must pay for this sales argument will be far too high.

At press time John L., his 400,000 miners, and most of 138,000,000 affected and largely indignant Americans were awaiting a court ruling on contempt proceedings and the applicability of the Norris-LaGuardia Anti-injunction Act as it applies to Government-operated properties. Recalling that in the past, John L. has always been "within his legal rights" in all his strike actions—no matter how disturbing they might be to the public, or how embarrassing to the Administration—the outcome of his current court battle was definitely in doubt.

Throughout Washington, however, the consensus was that "This is it—the too-long deferred showdown between labor and the Government's policy of labor expediency." It was admitted that John L. might win a battle but lose a war; that public resentment against such annual displays of the power of one man to cripple a nation's economy will doubtless result in Congressional tempering of the laws that have heretofore permitted John L. to be legally right regardless of the best interests of 140,000,000 people.

Action Urged

At the height of the coal crisis, many were urging President Truman to call a special session of Congress to deal with the situation in the event the strike were prolonged. Administration leaders were opposed, preferring to have the new 80th Congress—Republican dominated—call the plays. It was thought that action by a special session of the lame duck 79th Congress might prove too drastic and punitive to be practical, whereas members of the incoming Congress may be depended upon for quick and more realistic revisions of some of the one-sided and strife-producing labor laws now on the books.

Proposals for additional legislation are varied, but there appears to be considerable support for:

Major revisions of the Wagner Act to remove its bias toward organized labor.

Making unions subject to anti-trust laws; banning industry-wide bargaining.

Out-lawing the closed shop—considered a major factor in creating huge unions like the United Mine Workers.

Prohibiting coercion of individual workers by violence, through mass picketing.

Providing a new procedure to deal with strikes in public utilities and industries vital to the national welfare.

Banning secondary boycotts, imposing mutual responsibility on employers and unions to carry out agreements, and banning unionization of foremen.

Says G. O. P. Will Act

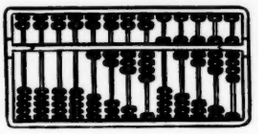
Recalling that much of the labor legislation passed during the last 12 years has been written with an eye to coming elections, the question has been raised as to whether Republicans of the incoming Congress will react similarly. Senator Ball (R., Minn.) a Republican leader who has been active in drafting labor legislation, says the Republicans will not dodge labor legislation for fear it might hurt them in the 1948 Presidential election.

"Republicans," he explains, "largely drafted and overwhelmingly supported the Case bill in the 79th Congress. If now, with power as well as responsibility, we ducked this No. 1 domestic issue, we could convict ourselves of the most blatant kind of political expediency."

President Truman, too, is expected to sign—instead of vetoing, as in the past—the type of legislation expected from Congress. Like the public, he is tired, it appears, of realizing that the only law under which the Administration could proceed against Mr. Lewis is the Smith-Connally Act, a wartime measure which is still in effect only because the end of the war has not been officially declared. This law prohibits anyone from inciting a strike against a Government-operated plant. Its provisions could not be adapted to any future strikes in steel, automotive or other basic industries —strikes which were considered as certain if the coal strike is successful.

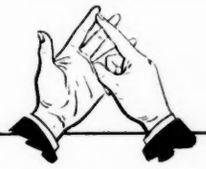
Vets' Housing Program

Housing Expediter Wilson Wyatt had a heart-to-heart talk with President Truman about the veterans' housing program on the day before Thanksgiving, but an expected decision on whether to continue or junk the program was not forthcoming. Doubtless the coal

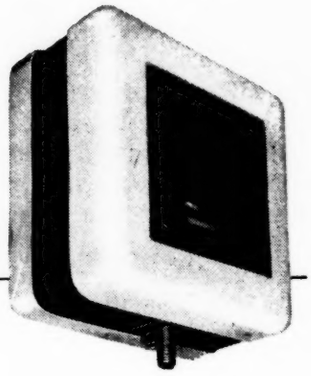


Whether you use an abacus

or count on your fingers



It's easy to add up the reasons for selling American Blower Unit Heaters



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2. Complete market coverage. We build 3 types of unit heaters: (a) Venturafin Horizontal Type for general industrial use—(2) Vertical Units for ceiling application—(c) Centrifugal Type for large hard-to-heat areas. Whether you are selling a giant industrial or a small store, you'll find the right type and capacity to meet your requirements in the American Blower line.

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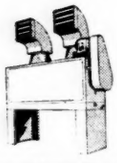
Venturafin Horizontal Type Unit Heaters



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strike and its probable effect on production and shipment of building materials had some bearing on the subject. At any rate, Wyatt was apparently told to let things ride along as they are until related circumstances clarify.

Situation in housing is that although certain phases of the Wyatt program—particularly those designed to increase production of building materials and equipment—have been quite successful, the program is considerably behind its goals of both started and completed residential projects.

The downward trend in new construction activity which began in September, continued in October, according to a November 26th report issued by the Civilian Production Administration. For the first time since the March 1945 low of $29 million, the uninterrupted upward trend in residential construction was halted, and this category declined slightly to $380 million in October, from the September peak of $385 million. However, in relation to total construction put in place in October ($1,036 million), residential construction represented almost 37 percent. At the end of the war it represented only 17 percent of the total.

In non-residential construction, the October volume of $656 million represented a drop of almost 6 percent from the August high of $694 million. Through the first nine months of 1946, non-residential building totalled $4,663 million, or within 3 percent of the $4,540 million made in August by the Inter-Agency Technical Committee for Construction Estimates.

Materials Up

Offsetting the downward trend of construction put in place—and responsible for the growing criticism of the Wyatt program favoring veterans—is the fact that many previously scarce items have reached a stage of supply and demand balance, while others are nearing that point. By the end of 1946, an overall cumulative supply-requirements balance is in prospect for brick, concrete block, cement, hot water heaters, warm air furnaces, and floor and wall furnaces. By the end of the first quarter of 1947, CPA looks for a cumulative balance in structural clay tile, asphalt roofing, sinks and radiation. It is considered likely that estimated requirements for clay sewer pipe will be met by the end of next spring, and that by the end of summer building boards will no longer be in critical supply. These estimates do not take into consideration the possibility of a prolonged coal strike.

At current and prospective production rates, no cumulative supply-requirements balance is in sight either this year or next year for cast iron soil pipe, bathtubs, lavatories, water closet bowls or insect screen cloth.

Maldistribution Blamed

Even in cases where overall production appears adequate in relation to existing requirements, CPA admits, local reports of delivery delays have been received. Reported shortages indicate a maldistribution which producers should be able to improve.

The building industry concedes this maldistribution, but blames the Wyatt program requiring set-asides for veterans' priority purchases. Industry contends that huge quantities of building materials, lumber and equipment are piling up in retailers' inventories—reserved for veterans' priority orders that do not develop. Many critics of the Wyatt program contend that if non-veterans were permitted to build freely, this backed-up material would be released to those who are in a position to use it. This, they contend, would permit builders of these non-veteran homes to move into new quarters, vacating their present quarters for use by veterans.

Bearing out this contention to some extent is the CPA report that for lumber generally the prospect of achieving a balance between supply and demand at an early date continues bright. In most yard stock items, CPA reports, such a balance has already been struck. September output of 3.2 billion board feet exceeded expectations by 200 million feet, and was 44% above September a year ago. Further rebuilding of mill and distributors' stocks may be possible in the near future on the basis of the latest revision of requirement estimates for 1946. Exclusive of pipeline needs, total demand on the basis of latest housing schedule and the current restrictions under the Veterans Housing Program Order, is now estimated to be around 32.5 billion feet whereas supply including imports seems certain to exceed 34 billion feet this year. If these estimates are correct, the lumber industry will be much better prepared to handle the expanded building program now scheduled for 1947.

With a view to eliminating some of the admitted maldistribution of building materials, CPA is reviewing Schedule B to Priorities Regulation 33 with an eye to an early revision of the set-aside provisions. These set-asides were introduced on September 1 as an emergency measure to speed up the completion of homes before the coming of winter.

Status of Controls

Rents—Expected to survive the Congressional ax, but with regional increases varying between 10% and 15% considered likely. Removal of rental and price ceilings on new homes considered likely.

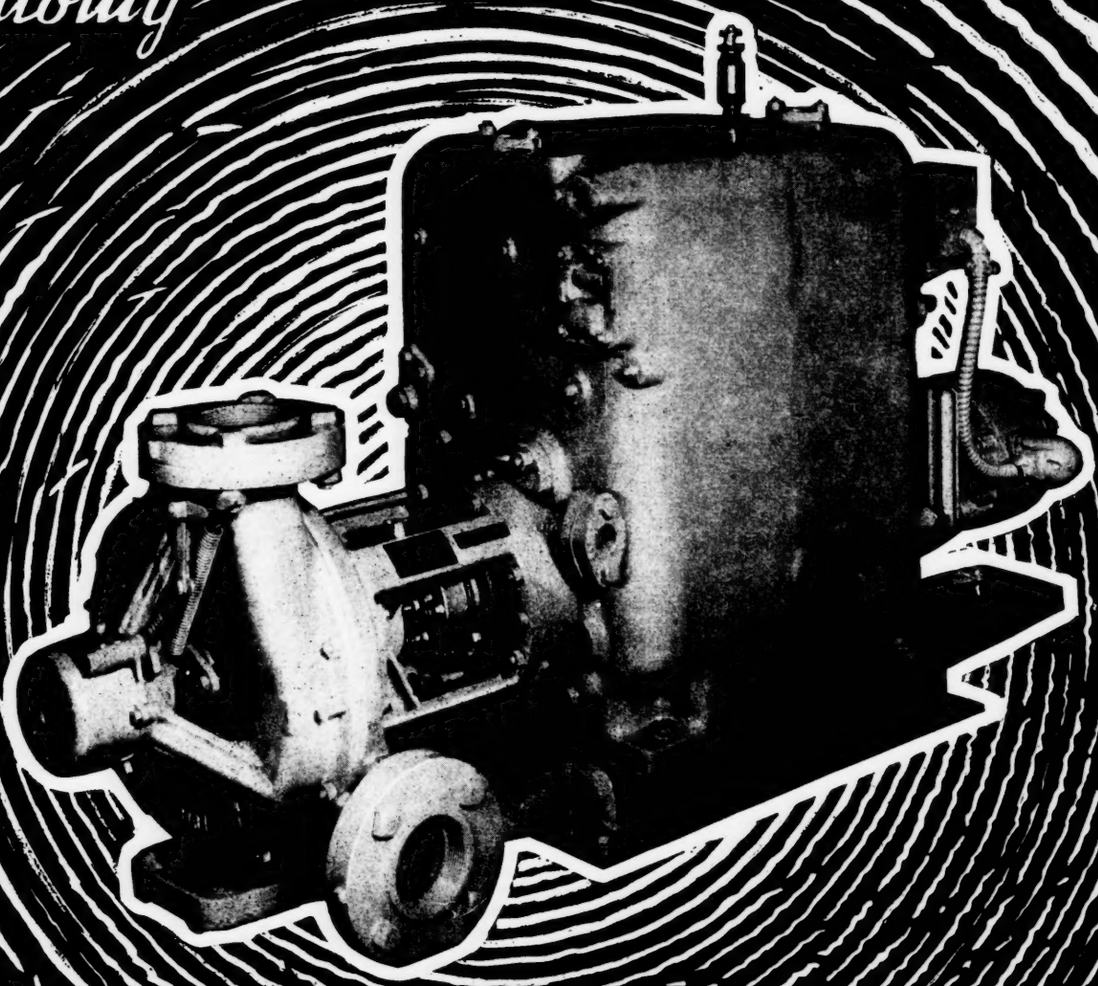
CPA Controls—Almost all of CPA's present functions are devoted to the housing program. If this program continues, priority powers and administration of non-housing construction order will probably be transferred to National Housing Administration.

Allocations over rubber, tin, lead, etc., will probably be transferred to Reconstruction Finance Corporation.

Credit Rules — Expect restrictions on installment buying to be limited to about 15 items instead of 40, with a floor, possibly $50, below which there will be no controls. Radios, automobiles, refrigerators and washing machines are expected to continue under controls, but jewelry and soft goods, charge accounts and single payment loans are likely to be freed.

Export Licenses—Most of the 900 items now under control to be dropped by January 1. Tin, lead, copper, building materials, tin plate, galvanized sheet and cold rolled sheet, zinc, fertilizer materials, trucks, tractors, tires, automobiles and batteries to be controlled for some time.

Centralized Heating Economy



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More important, this pump insures absolutely uniform circulation in system. That means continuous steam economy.

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THE NASH ENGINEERING COMPANY
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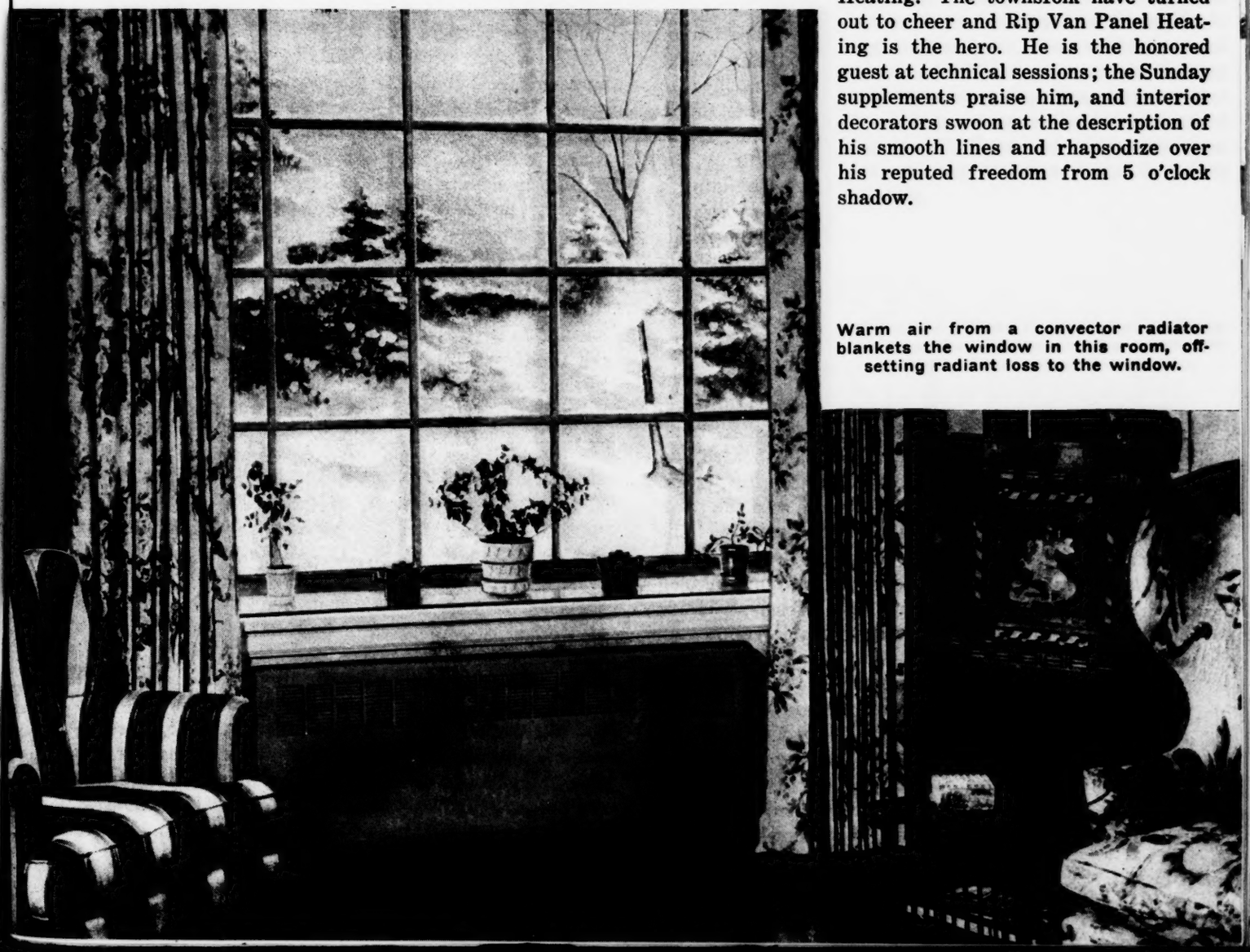
Will Radiant Heating Displace Convection Heating?

A. G. DIXON

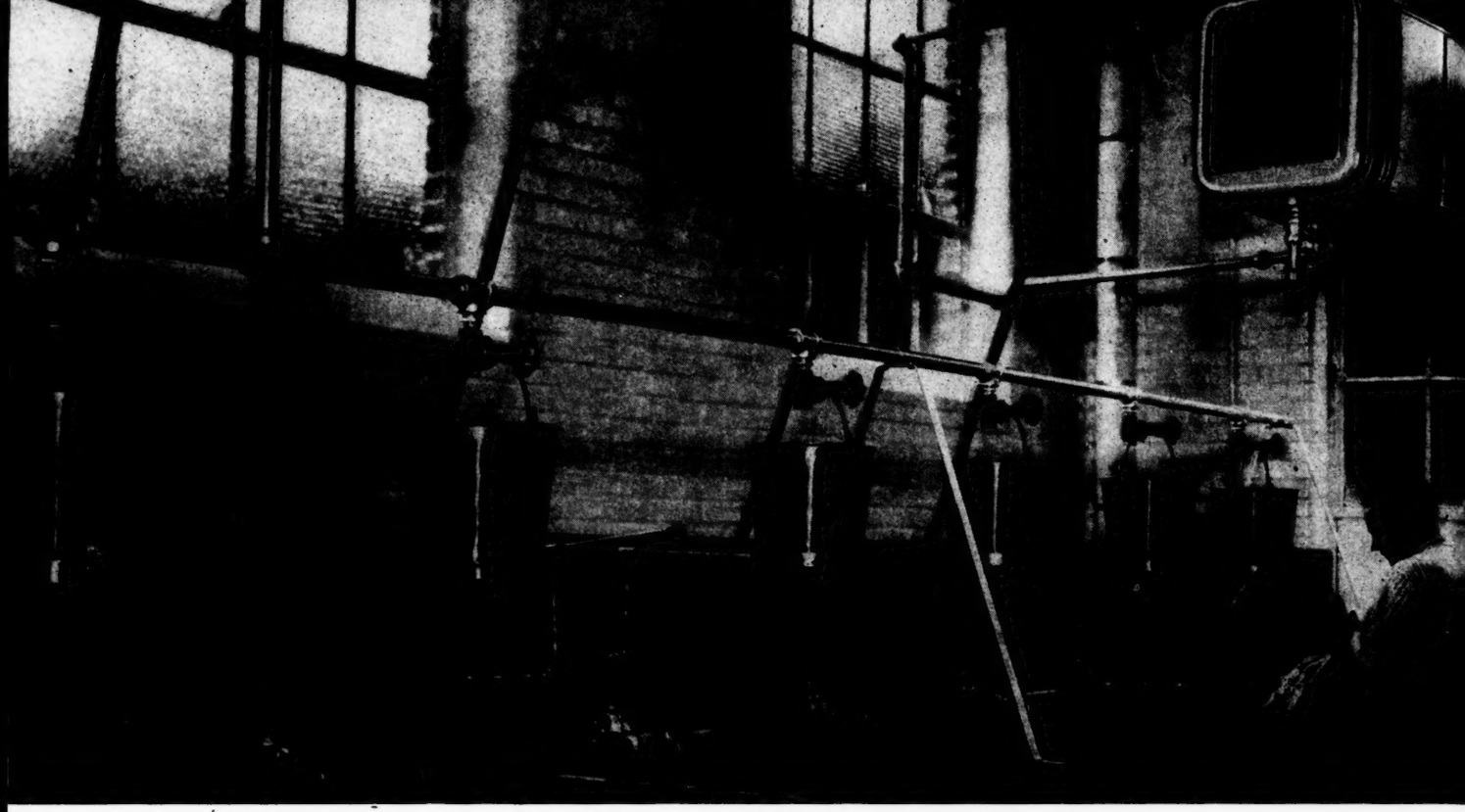
Secretary, Modine Manufacturing Co., Racine, Wisconsin

As pointed out in the accompanying article, Americans are extremists. We tend to exploit to the limit any new development (as we did with certain "miracle" drugs) and later discover with surprise there is room for both the old and the new. Will this happen with radiant heating and convection heating? Mr. Dixon presents here his reasons for thinking why it will. A champion of convection heating, he has here carried the offensive into the radiant heating camp with considerable spirit; in a subsequent article he will present more thoroughly the case for convection heating. Discussion on this controversial subject will be welcomed by the author and the editors.

RADIANT heating (or panel heating) is as old as the hills; and, as did Rip Van Winkle who disappeared into those hills long ago, it has come back to the haunts of man after a long absence. But whereas the old Rip of Irving's story was anything but a hero to his townsfolk, and whereas the dog that looked like, and probably was, a descendant of Rip's dog Wolf who snapped at Rip's heels when the long-whiskered oldster returned to civilization, it is not so with Rip Van Panel Heating. The townsfolk have turned out to cheer and Rip Van Panel Heating is the hero. He is the honored guest at technical sessions; the Sunday supplements praise him, and interior decorators swoon at the description of his smooth lines and rhapsodize over his reputed freedom from 5 o'clock shadow.



Warm air from a convector radiator blankets the window in this room, offsetting radiant loss to the window.



Unit heaters, which heat by practically pure convection, are a favored means for heating factories and similar space. Their ratio of weight to capacity is low, hence they respond quickly to varying demands, and this promotes economy of operation which is especially marked in buildings used only a part of the 24-hr day.

Twenty centuries ago the Romans arranged some of their buildings in occupied England and Germany in such manner that ducts built into the masonry of floors and walls carried the gases of combustion of charcoal fires outdoors by circuitous routing. These Romans, who were good engineers, thus applied to advantage two principles of heating. For one thing, they countered the too-much-then-too-little result which was normal to uncontrolled direct firing, by storing up in the mass of masonry of the building a residue of heat which was then slowly released. Second, they offset the chilling effect of uninsulated stone walls by causing the masonry adjacent to the flues or ducts to give off a compensating radiant warmth.

Rediscovery

It is interesting that the "rediscovery" of radiant heating was again associated with residual heating. A Mr. Barker of England, several decades back, observed that of two rooms, both centrally-heated and presumably equally heated, the one containing within itself an exposure of warm flue surfaces was the more comfortable. Again residual heat, tempering the fluctuations of direct firing in the absence of an adequate thermostatic control of combustion, combined favorably with radiant heating effect from an inside chimney wall to make a room comfortable for human occupancy.

To his credit, Mr. Barker was not only a keen observer, but also an aggressive engineer. He set about deliberately to reproduce this phenomenon which he had observed and which he had found good, putting pipes into the walls of rooms, connecting these pipes to a hot water heating system, and so siring the modern radiant heating installation.

Within a span of thirty years, up to the beginning of World War II, close to a thousand installations of radiant heating are reported to have been made in continental Europe and Great Britain. As might be supposed, there were troubles, of both structural and performance kind. The one sometimes tended to engender the other, as in some rather fine apartments in Paris where, in an effort to economize, an undue concentration of heat was applied in limited area ceiling panels adjacent to large window exposures, literally producing headaches among tenants who expcsed themselves for more than a short while to these exceedingly radiant panels; this is a not uncommon source of trouble in panel heating installations. Britain also has had her share of figurative headaches with radiant heating and both there and on the Continent there is strong inclination to continue to make haste slowly with radiant heating, employing it only selectively and in many instances jointly with convection heating.

A Slow Start

Meanwhile, in the United States, radiant heating got off to a slow start. It is true that about 1930 ripples were stirred in American heating circles when the then new British Embassy in Washington was completely radiant heated. But that ripple and others like it were submerged during the thirties by the tidal wave of popularity of air conditioning. The American people ate up air conditioning and the fad for it led naturally to exaggerations. Every furnace became an "air conditioner" and merchants sold "air-conditioned" shoes. The rambling heterogeneous air conditioning industry had a million dollars invested in it for each quarter of a million that should have

gone into its capital structure and the discordant voices at the Tower of Babel were in sweet harmony by contrast with the divergent claims of the ballyhoo which emanated from air conditioning's various schools of thought. The buyers, as might have been expected, caught it in the neck.

Some Kept Their Heads

Fortunately there were conservative engineers who "kept their heads while those about were losing theirs," and sanity took over, with the consequence that we came up to World War II well prepared to equip war production factories and the vehicles and ships of war with balanced air conditioning and convection heating systems that by maintaining freshened, cleaned air and controlled temperature in airplane cabin and ship's hold and blackout factory, lessened the physical strains upon fighting men and factory workers, thus aiding them in applying their full physical and mental abilities to their winning of the war.

We are a nation of extremes. It is recalled that Westbrook Pegler one time commented upon a visit to this country of a representative of French wine interests whose purpose was to promote the more extensive usage of wine by the American people. Pegler, although noting the representative's pleasing personality and evident sales ability, nevertheless claimed that an attempt to convert Americans from whiskey to wine would be futile because, he explained, "When an American wants to get drunk, he wants to get drunk in a hurry." It seems we may be in a similar extreme position as to radiant heating—not temperate, but with the disposition, "If it's any good let's drink the whole bottle."

Cool Analysis Needed

In the writer's opinion there is cause for concern that the issues are no longer confined to cool, analytical discussions among architects, heating engineers, and heating contractors. The contractor, the engineer, and the architect are already being kept so busy giving clients the reasons for employing panel heating with discretion as to be handicapped in efforts to educate themselves further about these very precautions which they have voiced. They are already meeting with such a degree of bland assumption on the part of customers fortified with hot cash that of course panel heating is the thing for their new building, as to be tempted to go along, with a shrug, on the theory that the customer is right if he is in position to pay for what he wants. How did we get this way? How did a large fraction of the people of a great nation get so quickly indoctrinated, and what will be the regrets that will follow this gulping of something that should be carefully chewed in small portions if a new case of indigestion is not to parallel the one experienced in air conditioning a decade ago?

The present radiant heating situation indeed bears a marked resemblance to that in air conditioning over a decade ago. In the period 1940-45 we had a war-enforced holiday in residential building, simulating the depression-spawned one of 1930-35. During both these periods of building drouth, the shelter and home magazines seized upon any interesting new form of copy and played it up to their readers, in substitution for the customary generous supply of factual data drawn from actual house building operations. The readers, awaiting better times in the earlier period, and in the latter instance awaiting the end of direct war interferences with home building, avidly soaked up the dissertations on "modern radiant heating." In what we may hope was the zenith of the hysteria, a leading picture weekly presented a "scientific" summary of radiant heating which included the profound commentary, "The air in the house stays cool. This saves money because air is hard to heat." Up to now there seems to have been no Sunday supplement proposal that atomic energy be employed as the direct primary source in combination with radiant panel heating, but this may be expected any day.

Men at Work

Once more there are hard at work — praise be! — uneffusive and realistic engineers, scientists, and contractors, who are analyzing and testing radiant heating theories and methods, just as they performed this same quiet service during the bombast period of air conditioning a decade ago. They have nothing to sell; or if they have, they subordinate fancy to fact and advise potential customers what is good and what is not wholly good and what is not yet wholly understood as to its degree of goodness or badness, about the application of radiant heating principles.

From these sincere efforts will come broad benefits. There can be no fair doubt of the existence of substantial values, available to us if we properly exploit the principles of radiant heating, although the field for particular and specialized application of these principles is infinitely more narrow than is the case with air conditioning.

Let us examine, in the simplest possible terms, the claims which are put forward for panel heating. Let it be agreed that we are all reasonably informed and reasonably persuaded that human dissipation of self-generated heat is most satisfactorily accomplished if the normal shares of about 50% radiative, 30% convective and 20% evaporative-respirative body heat losses (sedentary) are reasonably maintained, and provided that any absolute increase in one of these loss factors is suitably compensated by a diminution of another of these loss factors.

Comfort

If we agree on these fundamentals, then, drawing one's conclusions directly from claims for panel heating, it is apparent that by far the greatest single argument advanced for panel heating is that it promotes human comfort. We are informed that a cold outside wall "saps" the radiant heat from a man; we learn, however, that a man has several sides to him and that if another of the areas to which the man is

exposed is made a little warmer than normal (a floor, a ceiling or another wall) then the *man's* total radiant output, or loss, is kept at such a point that he feels comfortable. This is of course over-simplified, but in bare outline it expresses the principal reason for being, of radiant panel heating.

There is no fallacy whatever in this reasoning. Fallacy begins to enter only when we proceed, as every engineer must, to examine the general soundness and the general economy of the application of a theory which he has already to his satisfaction determined is correct as a theory.

The simplest way to state the fault that has been allowed to enter the application of the completely sound theory expressed in the second preceding paragraph, is by an inverse procedure: *To accomplish the wholly laudable objective outlined in the second preceding paragraph, in an estimated 95% of spaces built for human occupancy, the architect or engineer would have to do only what he would do anyway in the modern construction of a room; that is to say, insulate well all exposed areas. Thereupon he would need only to introduce heat to the room by the most convenient means and the most economical means,* in sufficient amount to offset the building heat losses that are inevitable regardless of the extent of insulation.

The Normal Room

In the normal room an indiscernible difference in human comfort—comfort expressed in terms of maintenance of a sound and proper relationship between radiative and convective heat losses from the body—results from the use of convection vs. radiant panel methods of introducing to such a room the heat required to compensate losses to the out-of-doors. (A "normal room" does not here include an open-window tubercular hospital room, nor a room in which there has been employed none of the advantages available in modern insulating materials and technique.)

The writer is not in the least original in these statements. The following is quoted from the 1946 *Guide* of the American Society of Heating and Ventilating Engineers: "In general, radiant heating of occupied spaces is not a procedure designed to create differences between air and walls, but is merely one method of introducing heat into that space," and "In practice, where radiant heat is introduced into a room, that heat is absorbed by surfaces, furniture, and the like, and then transformed into convective heat so that air and surfaces tend to attain a generally uniform temperature."

Insulation

Also, from a keen and hard-headed engineer, Walter L. Fleisher of New York City, a former president of the ASHVE, we have this: "It does not make any difference how radiation is produced as long as the surrounding walls are sufficiently warm so that the radiation from the body to the cooler surfaces is not too great for the metabolism of the body to keep pace with it.—Of course we are approaching radiant heating more and more, with the increasing insulation of our buildings.—*The whole tendency toward better construction, the better insulated construction that is used today, makes many of our houses, no matter how they are heated, panel-heated houses.*"

Recapitulation

This vein of reasoning is not unduly complex but perhaps being unfamiliar, it may be benefited by a recapitulation; it is well at times to review any reasoning in terms of simplest fundamentals. Restating the foregoing, therefore:

(1) Convection heating of a room causes the warming of furniture and walls which then radiate that warmth to the human body and, conversely, radiant panels warm opposed walls and furniture, which in turn transmit their heat to the room air. This of course merely follows the primary law of flow of heat from warmer to cooler bodies;

(2) Our human abode "heating systems" are in reality cooling systems for the human body, inasmuch as the body generates heat constantly and must as constantly dissipate the excess of such heat not required to keep the blood at about 98F and the skin at about 90F. Thus the sought-after winter room temperature of approximately 70F serves deliberately to provide controlled cooling for the body;

(3) Consequently, it is not essential that any of the room surfaces to which our bodies are exposed should be above 70F. It is in fact only the difference between their temperature and the temperature of the *body surfaces,* that enables the sedentary human body to lose approximately 50% of its excess heat by radiation;

(4) To guard against misunderstanding resulting from repeated reference to sedentary conditions, it should be recalled that as the body becomes more active it throws off a greater and greater share of its excess heat in the form of latent heat, and that our concern with radiant losses from the body lessens accordingly;

(5) Just as a man exposed to excessive velocity of air movement experiences undue convective and evaporative heat losses that need to be balanced out by the addition of several degrees to the temperature of such air if the man is to be comfortable, so a man exposed to an excessively large area of quite cold room surface experiences undue radiative loss that must be balanced out by the addition of several degrees to the temperature of another of the room's surfaces if the man is to be comfortable;

(6) But the normal room of current construction does not have excessively large areas of quite cold surfaces. The normal room of today has a mean radiant temperature within a very few degrees of 70F when the room itself is at 70F.

And as implied above, radiative and convective heat inevitably merge and *only a high rate of heat loss from a room can prevent this merging and transmutation*

A pipe-in-masonry panel heating installation in process. Massiveness and high residual heat are its components, requiring extraordinary means for temperature control of the heating medium.

from wiping out distinguishability of the origin of the heat received by the room, whether convective or radiative; and

(7) Hence, with the exception of the decidedly abnormal room, some reason or reasons other than that of serving human comfort better must be advanced by the proponents of panel heating in order to justify the selection of this method as contrasted with convection means of introducing heat to a room.

Responsiveness

Still striving for the utmost simplicity of analysis, let us consider another of the principal claims of panel heating enthusiasts. It is stated that panel heating is smooth, even heating; that out-door temperature fluctuations are not felt by the occupants of a panel heated room. Let's take this one apart.

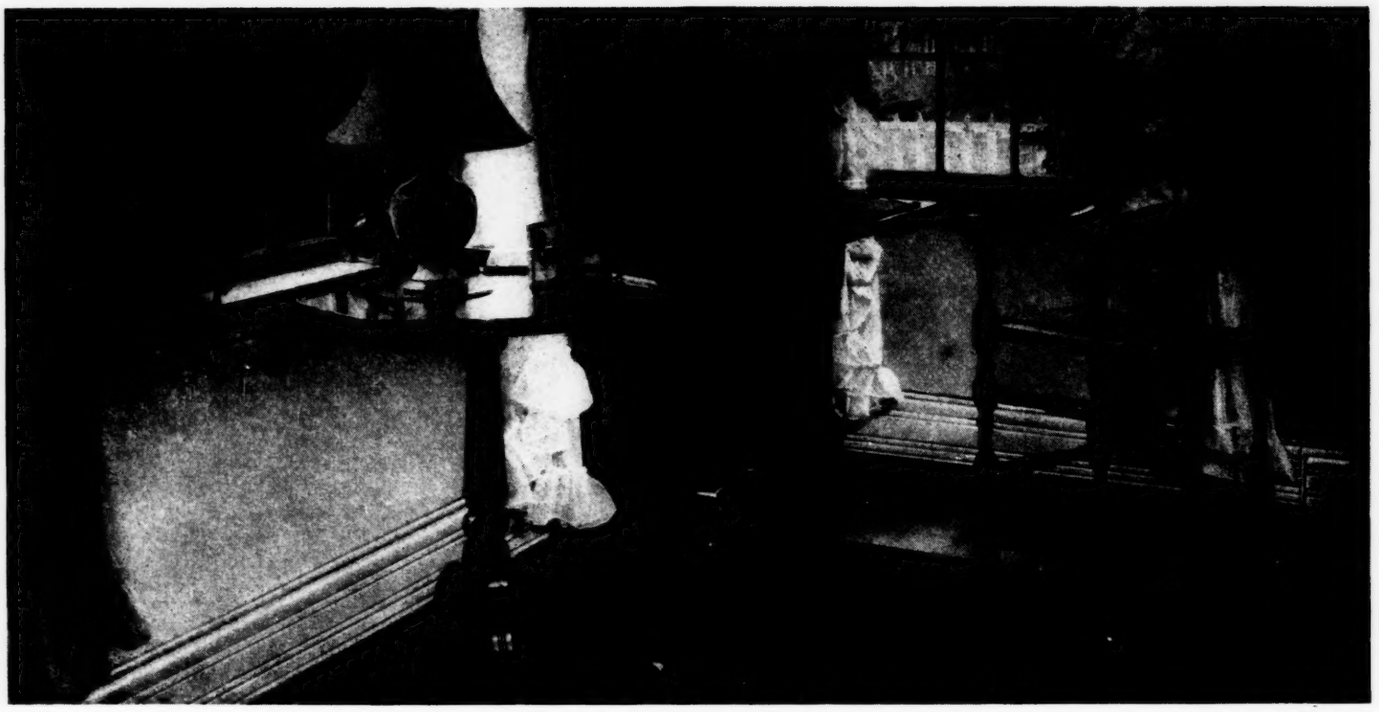
The fact is that a panel heating installation, particularly the type where tons of pipe are buried in additional tons of masonry, epitomizes sluggishness of response. A deliberate return has been made to the high ratio of residual heat, whence the profession and the industry has been departing for the past twenty years. At a time when stepped-up performance of our railroad trains, our automobiles, our lawn mowers and practically everything else mechanical is being accomplished, when ounces of clothing serve where pounds were once required and when even a favorite argument for structural materials today is their lightness of weight, we are proposing to multiply many-fold the mass or poundage of that portion of our heating system which dissipates or transfers heat to our rooms. Actually, one of the freely recognized problems of panel heating installation is that of controlling the heating medium—careful and anticipatory controlling of its temperature, and controlling of its rate of flow —in order to compensate for the sluggishness of response inherent in this method of heating. It is a curious and incongruous although incidental point that the exponents of panel heating urge heavy usage of insulation; this because the problem of slow adjusting of a panel heating surface to fluctuating demands is thereby tempered.

To put it briefly, panel heating can be satisfactory only with the employment of the latest and best developments as follows:

(a) Some manner of thermostatic control of a highly anticipatory kind, preferably including an out-door thermostat;
(b) Means connected thereto for modulating the temperature of the heating medium, and
(c) Means likewise connected to such thermostatic control, to provide continuous or approximately continuous delivery of the heating medium.

Modulated Heat Flow

Now, if we should favor a convector or radiator heating system, or a winter air conditioning system, with even a portion of these same advantages of continuous flow of modulated heating medium, we would note a distinct improvement in the performance of such a convection heating system; the "cold 70" which follows upon intermittent or cycling operation of a system would be entirely eliminated. As a matter of fact, balanced continuous delivery of modulated heat is simpler to effect with a convection system than with a panel system, for there is not to be taken into account in the convector system the tremendous lag which can be properly dealt with in a panel system only by employing the most elaborate of control auxiliaries.



The answer of the cast iron radiator industry to panel heating demands—the radiant baseboard or radiant baseboard convector. Photo courtesy Burnham Boiler Corp.

Certainly a fair comparison between the performance of a convection and that of a panel heating system can be made only if such advantage as may accrue from the use of high grade auxiliaries *is extended to both systems alike.* A favorite American diversion is to compare the performance of one's automobile with that of a friend's automobile, but to employ a parallel to the above, not even the warmest of friendly competition would prompt a man to argue that he should have 80 octane gasoline in his car but that his friend should be permitted only 68 octane gas.

A Complex Subject

The subject of radiant panel heating, and of the analysis of its aspects set alongside those of convection heating, is far too complex and involves too many conflicting claims to be analyzed in a short article such as this. However, in this quick review of panel heating claims, methods and results, we would be falling far short of completeness if we did not briefly consider the major accommodations that are necessary in the architectural and construction treatment of a building to meet the special demands imposed by panel heating—accommodations especially needed with the pipe-in-masonry form of panel heating.

In the present state of the art it is almost necessary to build a house around its panel heating system. Special considerations where pipe is to be interred in building masonry, with the earnest prayer that it may never again see the light of day, are of such impelling strength as literally to dominate many structural aspects that in the instance of a convector heating system are practically independent of the heating plant. It is to be reasonably questioned whether panel heating virtues are of such high and unique character as to deserve favors of this degree. Finally, panel heating trespasses on the perennial fundamental of economy with quality, for it is idle to claim that currently a panel heating system is as economical to install as is a convection system, taking all factors of respective completed installations into account.

Enthusiasm is a splendid quality, within bounds. Enthusiasm with respect to panel heating is serving certain good purposes, such as accelerating our acquiring a basic understanding of the whole of the problem of comfort heating and such as promoting the commercial availability of better comfort heating control methods, for these latter will pay off in improved performance of convection as well as panel heating systems.

Sheep from Goats

As this is written, it would be difficult to put a finger upon the individuals or organizations who were most responsible for the screw-ball period through which air conditioning passed ten years ago, and from which it has not yet completely recovered in terms of public understanding and public good will, but it is easy to put to the test of common sense the exaggerations which are being projected in the name of radiant heating. There are some sheep in the flock but there are a lot of goats, too; let's separate them.

As bearing upon the question raised by the title of this paper, and should my opinion at this point be still in the least disguised, I do not believe that radiant heating will displace convection heating.

In a subsequent issue, Mr. Dixon will summarize and evaluate the advantages of convection heating.

Is Air Conditioning Profitable in Industry?

NATHAN N. WOLPERT
Associate Editor, HEATING AND VENTILATING

Did you ever wonder whether air conditioning in industry warranted its installation and operating costs? Here are some facts based on the experience of plant executives as shown by a recent survey.

HOW effective is air conditioning in industry? As with many common questions, most persons assume an answer based upon individual experience which in general cannot be substantiated by basic facts.

Therefore, HEATING AND VENTILATING sent letters to a number of key industrial plants that have had a chance to observe the results of air conditioning, particularly as against a time when air conditioning was not employed. Factories that functioned only as war plants were excluded.

The following questions were asked:

1. Has air conditioning increased the productivity of workers in your plant?
2. What has been employes' reactions to air conditioning in the way of absenteeism, employe turnover, efficiency and morale?
3. Has air conditioning decreased the amount of spoiled or sub-standard merchandise in your plant?
4. Has the gain in productiveness outweighed the cost of the air conditioning system?
5. What is the most important function of air conditioning in relation to your product in control of temperature, control of humidity, and control of cleanliness?
6. As far as your particular plant is concerned, are there any changes that you would suggest to improve your system?
7. Have you any comments?

While some supplied comments based on personal experience, in addition to the questions asked, most of the replies received were in direct answer to the quiries.

Of the replies received, 95.4% indicated that air conditioning had increased the productivity of workers.

Some engineers seemed reluctant to reply to the question regarding the effect of air conditioning on employes' reaction, because they said that intangible results had to be measured. However, of the many who replied, 65% reported that conditions were generally good and 10% said that there was no difference in results. Where the replies were more specific, 15% reported a reduction in the amount of absenteeism, and 5% no reduction; labor turnover—no reduction, 5%; increase in efficiency and morale, 20% each.

From one electrical plant came the report that because of air conditioning there was a marked improvement of employes' reactions in the office but not in the shop—in fact, a number of shop workers complained about colds. A few reported that conditions improved during the summer months.

The question regarding a reduction in sub-standard products brought a 100% reply that there was an improvement in quality.

A chocolate candy manufacturer said, "Most of our processes could not operate without air conditioning, therefore we have no choice." A pharmaceutical company reported that in many of their processes, air conditioning is a must. A publisher found that he had a 15% improvement, chiefly due to the control of temperature and humidity.

As to whether the gain in productiveness outweighed the cost of the air conditioning system, 88.6% replied yes and 11.4%, no.

In an effort to find out the importance of temperature, humidity and cleanliness control, 22.6% reported these factors were equally important. The most important is humidity control with 52.5% followed by temperature control 39%, and cleanliness, 26%. There is reason to believe that cleanliness control is becoming more appreciated as an air conditioning benefit.

An optical works reported that control of temperature was important in the prevention of perspiration on finished optical parts; humidity control was of minor importance within a reasonable range but that cleanliness control was very important in optical assembly work. A watch manufacturer stated that while all three factors were important, humidity and cleanliness control possibly outweighed temperature control.

A large publisher said that temperature control is important in offices; humidity control important where paper is handled; and control of cleanliness where dust spots may show on the film being handled. Said an official of a large research laboratory, "In laboratory work covering a wide range of research it is becoming more important to have control of atmospheric conditions."

The question for suggestions as to changes to improve the present system brought forth the following replies:

Automotive Plant: Would like to put windows in one plant so that people can look out.

Optical Works: We would consider the use of perforated ceilings for air distribution instead of air diffusers.

Electrical Manufacturer: More changes of air per hour in the shop.

Optical Works: Better air filter, improved draftless distribution, reduction in maintenance, and more reliable controls.

Machine Company: Would like to see maintenance charges reduced.

Optical Works: Better humidity control.

Candy Manufacturer: Improvements in type of grille on air outlets to prevent drafts.

Roller Bearing Plant: A better control of humidity and a better method of reheating the air.

Publisher: Better humidity control.

Pharmaceutical Company: Air conditioning set-ups that will give uniform dry and wet bulb control throughout the year seem best for our production.

The following replies were received in answer to Question 7 which invited comments.

Motor Accessories Plant: Air conditioning should not be combined with windows that cannot be opened. It should be so arranged that supplied air can be increased or decreased in offices whenever necessary and in proportion to the number of people in a room.

Candy Manufacturer: We find that air conditioning helps materially in the office as well as in the plant.

Food Canner: It has ceased to be an industrial luxury when applied properly.

Surgical Specialties: Many more could use it to profit and advantage.

Optical Works: We know the majority of workers enjoy and appreciate air conditioning.

Very interesting comments were received in a letter from an official of a publishing company who wrote:

"There is no doubt of the benefits to be derived from air conditioning in any plant. It is difficult to ascertain, however, what percentage of improvement is made as a result of installation of air conditioning. My guess would be about 15%, chiefly due to the control of temperature and humidity for a product such as ours—paper.

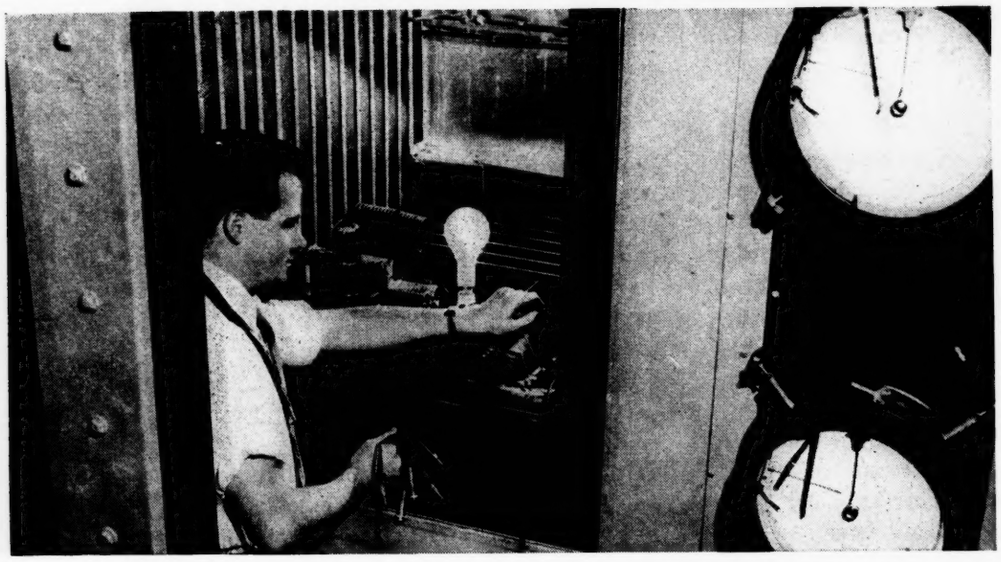
"I regret to say that the cost of air conditioning is still so high as to make it unprofitable to install it in all departments. I recently made a survey in the hope that the cost had been reduced but found it prohibitive. If and when the cost is reduced to a reasonable figure, we will be in the market for air conditioning a considerably larger area than that which we now use."

Another large publisher said, "Our air conditioning is limited to humidity control in areas where paper is stored or handled. By eliminating static electricity and keeping paper stock from drying out, work has been made easier in the press room and bindery."

There seems to be conclusive evidence that air conditioning is an industrial necessity—that it will cut down on the amount of sub-standard output and that the gain in production will also, over a nominal period, outweigh the cost of the air conditioning system. While at first glance one may believe that temperature control is of most importance in an air conditioning system, most plants find that the most important factor is humidity control. Air conditioning has advanced greatly from the days when it was a novelty and a luxury. Today, in most cases, it is an industrial necessity.

Test Chamber for Electronic Tubes

A tube testing chamber capable of producing temperatures from —100 to 170F was produced by the General Electric Co., for testing radio tubes. A variety of humidity and pressure conditions can also be produced in the chamber so that a wide range of altitude and climatic conditions can be duplicated.



Controlled temperature and pressure chamber for testing electronic tubes.

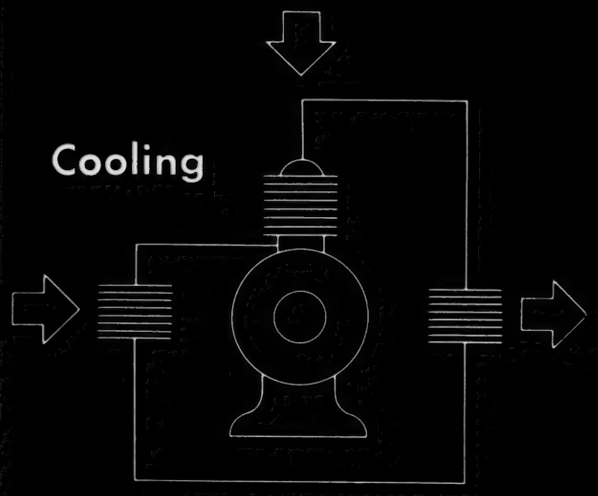
Two dials outside the door of the chamber maintain a 24-hour record of conditions inside the test box.

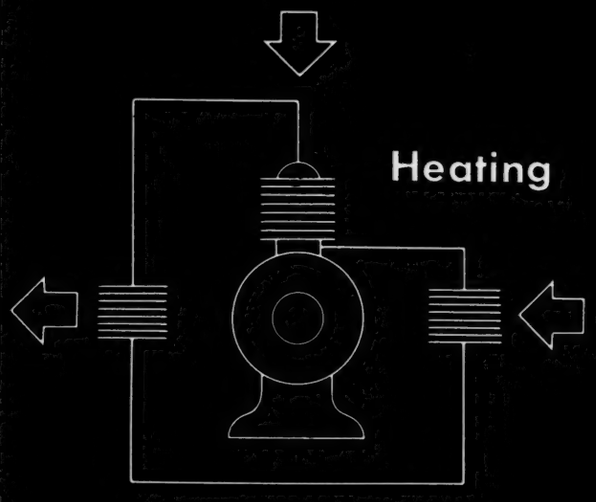
Elaborate refrigeration and pumping equipment makes possible quick change to any desired altitude and climatic conditions. The tube icebox "climbs" at a rate of 3,000 ft per min, reaching 80,000 ft in about two hours.

Refrigeration is accomplished from 70F to —100F in 2.5 hours. Refrigeration equipment consists of a 2-stage plant, the first compressor operated by a 30 hp motor and the second by a 20 hp motor.

—*F. N. Hollingsworth.*

Progres





A HEATING AND VENTILATING PROGRESS REPORT

PROGRESS REPORT ON THE HEAT PUMP

E. R. AMBROSE

Air Conditioning Engineer, American Gas and Electric Service Corp., New York, N. Y.

THE heat pump, frequently referred to erroneously as a reverse cycle refrigeration system, is a compression type refrigerating system consisting of evaporator, condenser, compressor and expansion valve. It is so arranged that it can be used to cool air in summer in the conventional way and, in addition, be used to supply the winter heat by pumping the heat removed from a low temperature level to a satisfactory higher temperature level.

In summer, the heat rejected or dissipated by the condenser enters the system from two principal sources, from the room air being cooled, and from the mechanical equivalent of the electrical input to the compressor. In winter, the heat pump makes available for heating the room, the heat equivalent of the electrical input as well as heat taken from the outside air, water or ground. Of these two heat components, the heat energy from an external source is obtained as a sort of by-product from expenditure of the electrical energy, so that we have the rather paradoxical case of paying for only a fraction of the heat input to the room.

If Q_1 is the refrigerating effect of a refrigerating system and Q_2 the energy input to the compressor motor to produce this effect, then the heat obtained from the heat pump is $Q_1 + Q_2$ and the ratio of total heat obtained to heat paid for is

$$\frac{Q_1 + Q_2}{Q_2}$$

This ratio is called the coefficient of performance, and with the heat pump ranges in actual practice from 3.0 to 5.5. Consequently, the cost of heating with a heat pump is about 18% to 30% of that with electric resistance heaters or other forms of electric heating.

Basic Types

There are a number of possible arrangements of a heat pump system, as shown in Fig. 1 to 6, inclusive. In air to air arrangement given by Fig. 1, during the heating cycle air is used as a source of heat, and air is used to remove heat from the condenser. During the cooling cycle air is used to cool the space and air is used to reject the heat to the outside. The path of the refrigerant is reversed by means of eight 2-way valves as shown.

During the cooling cycle, the gas refrigerant goes from the refrigerating compressor to surface *A* where it is liquefied, giving up heat to air outside the conditioned space. From surface *A* the liquid refrigerant goes to the liquid receiver, through the expansion valve to surface *B* where it is gasified, absorbing heat from the air being delivered to the conditioned space. From surface *B* the gas returns to the compressor.

During the heating cycle, the gas refrigerant goes from the compressor to surface *B* where it is liquefied, giving up heat to the air being delivered to the conditioned space. From surface *B* the liquid refrigerant goes to the liquid receiver, through the expansion valve to surface *A* where it is gasified, absorbing heat from the air outside of the conditioned space. From surface *A* the gas returns to the compressor.

Instead of reversing the direction of the refrigerant, as shown in Fig 1, an alternate design, Fig. 2, maintains a fixed refrigerant circuit and reverses air flow. During the cooling cycle, air from the conditioned space passes over the cooling coil, thence back to the conditioned space. Outside air passes over the condenser to the outside.

During the heating cycle, air from the conditioned space passes over the condenser, and back to the conditioned space. Outside air passes over the cooling coil, back to the outside.

Fig. 3 shows an air to liquid design. Air is used as a source of heat; liquid is employed to transfer the heat from condenser and chiller. In this design, the refrigerant circuit is fixed, going from compressor to condenser, through the expansion valve and chiller, back to the compressor.

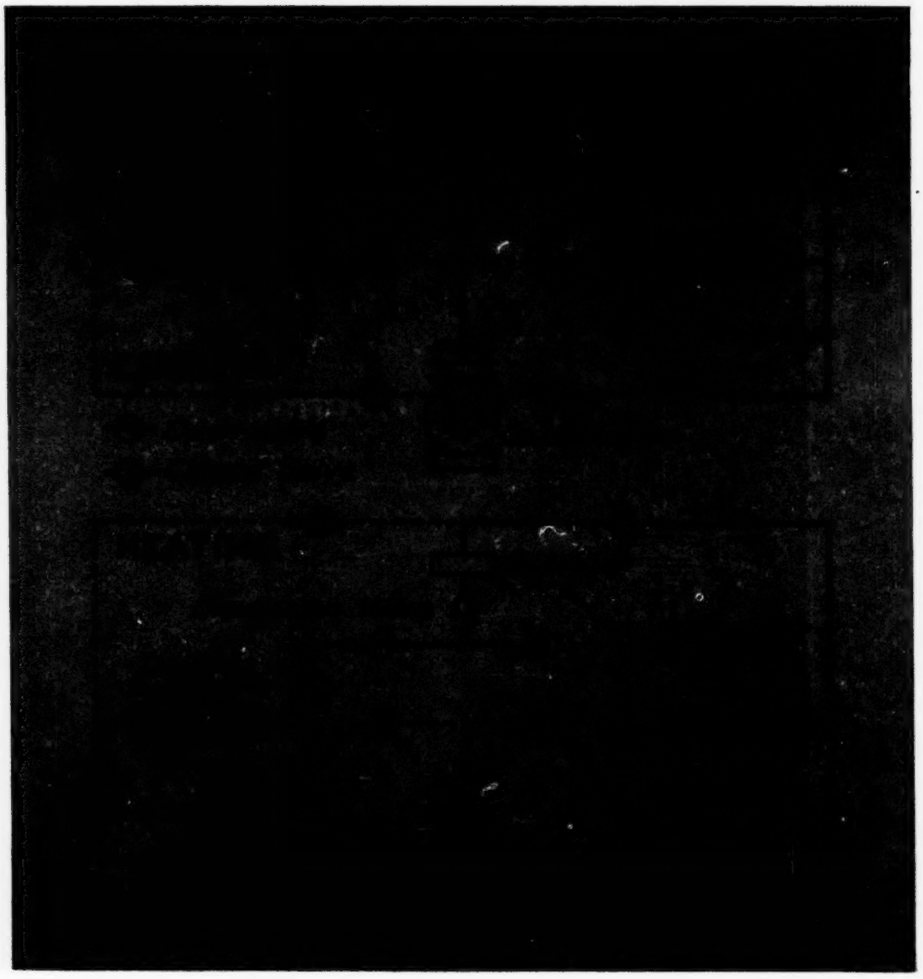


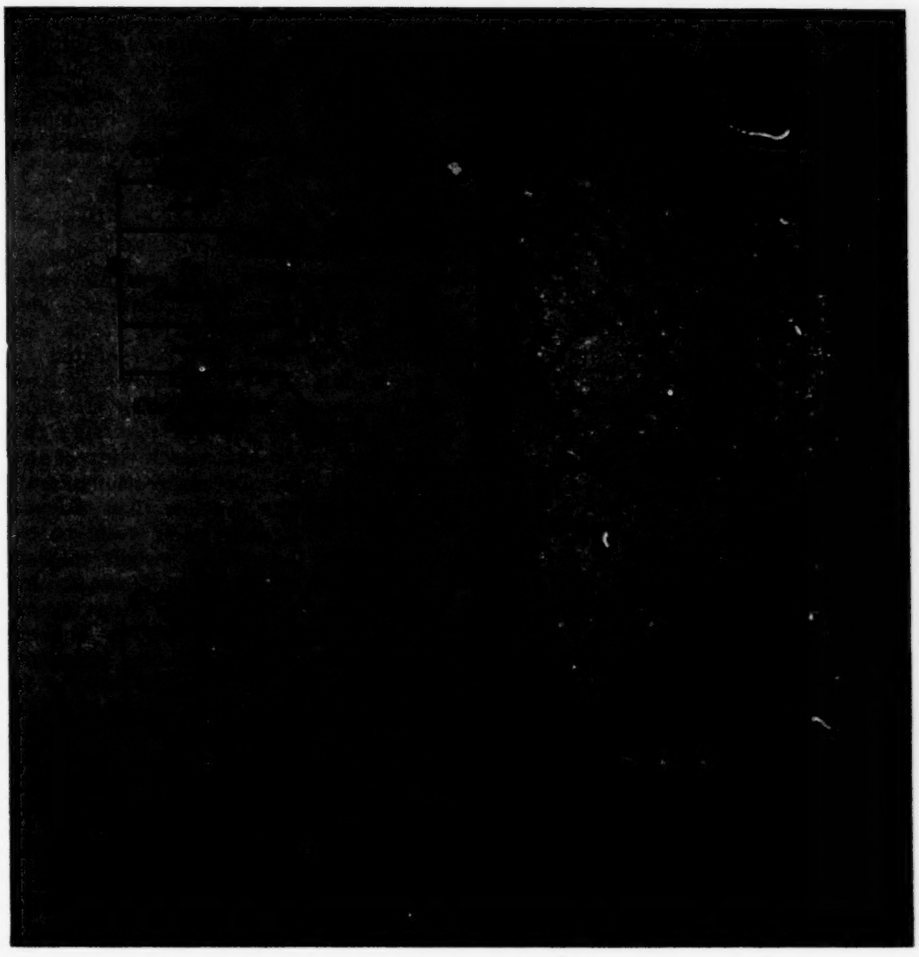
Fig. 1. Air-to-air design of heat pump. Solid lines show refrigerant path in each case, dotted line the unused path when performing function indicated. Refrigerant path is thus reversed by eight 2-way valves.

During the cooling cycle, a pump circulates the colder liquid through the chiller, the inside surface, then to the pump. The circulating liquid in passing through the inside surface absorbs heat from the air being delivered to the conditioned space and in turn gives up this heat to the refrigerant in the chiller; another pump circulates the warmer liquid through the condenser, the outside surface, then back to the pump. The liquid in passing through the condenser absorbs heat from the refrigerant and gives it up to the outside air when passing through the outside surface.

During the heating cycle, the pump circulates the colder liquid through the chiller, outside surface, then back to the pump. The liquid in passing through the outside surface absorbs heat from the outside air which is transferred to the refrigerant in the chiller. The other pump circulates the warmer liquid through the condenser, inside surface, then back to the pump. In passing through the condenser, the liquid absorbs heat from the refrigerant which is transferred to the conditioned air passing over the inside surface.

Fig. 4 shows a water to water design. Water is used as a source of heat, and water is used to transfer heat from the condenser and chiller. In this design, the refrigerant circuit is the same as in Fig. 3, going from the compressor to the condenser, through the expansion valve and chiller, back to the compressor.

Fig. 3. Air-to-liquid arrangement with air as source of heat and liquid to transfer heat from condenser and chiller. Solid lines show path of refrigerant and liquid when performing function indicated, dotted lines the unused paths when performing function indicated.



During the cooling cycle the deep-well pump circulates water from the well through the condenser, then to the drain. The well water absorbs heat from the refrigerant when passing through the condenser. Another pump circulates the water through the chiller, the inside surface back to the pump. The circulating water in passing through the inside surface absorbs heat from the air being delivered to the conditioned space and in turn gives up this heat to the refrigerant in the chiller.

During the heating cycle, the deep-well pump circulates water from the well through the chiller, then to the drain. The refrigerant absorbs heat from the water as it passes through the chiller. The other pump circulates the warmer water through the condenser, inside surface back to the pump. The water in passing through the condenser absorbs heat from the refrigerant which is transferred to the conditioned air passing over the inside surface.

Fig. 5 shows a water to air design. Water is also used as a heat source in this case, and air is used to transfer the heat from the condenser and chiller.

During the cooling cycle the gas re-

Fig. 2. Air-to-air design with fixed refrigerant circuit but air flow reversed by dampers.

frigerant goes from the compressor to surface *D* where it is liquefied, giving up its heat to the water from the deep well. From surface *D* the liquid refrigerant goes to the liquid receiver, through the expansion valve to the inside surface where it is gasified, absorbing heat from the air being delivered to the conditioned space. From the inside surface the gas returns to the compressor.

During the heating cycle the gas refrigerant goes from the compressor to the inside surface where it is liquefied, giving up heat to the air being delivered to the conditioned space. From the inside surface the liquid refrigerant goes to the liquid receiver, through the expansion valve to surface *D* where it is gasified, absorbing heat from the well water. From surface *D* the gas returns to the compressor.

Fig. 6 shows a practical and feasible method of heating hot water by means of a heat pump. With delivered hot water temperatures of 130 to 150F it is possible to obtain an average coefficient of performance of 2½ to 4. The heat source for a hot water heat pump can be ambient air or any other heat source which has been mentioned as suitable for the heat pump air conditioner. In addition to heating the water this unit will provide some cooling effect which

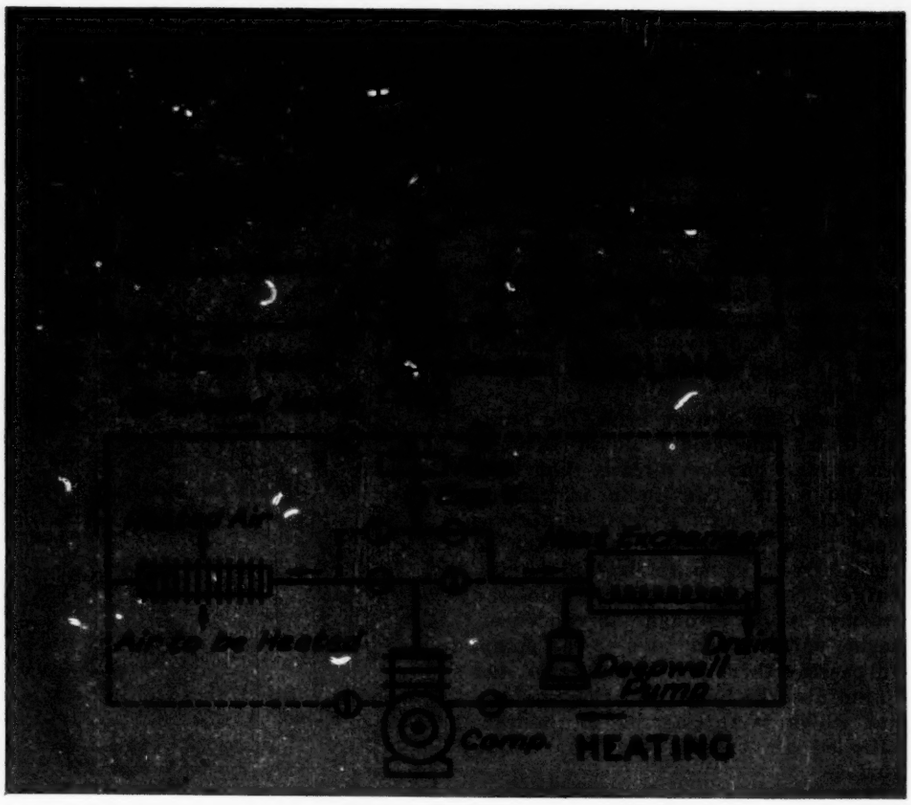


Fig. 5. Water-to-air heat pump with water as source of heat and air as medium to transfer heat from inside surfaces.

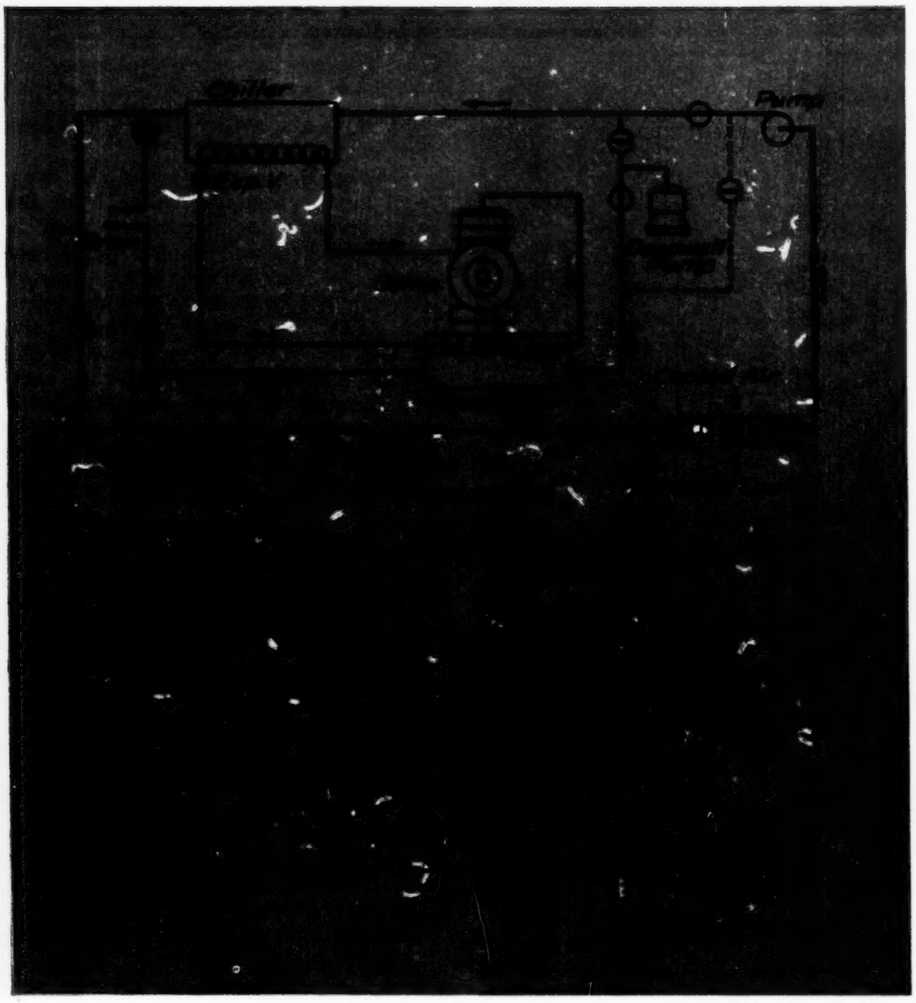


Fig. 4. Water-to-water heat pump with water as both source of heat and medium to transfer heat at condenser and chiller.

can be used to cool and dehumidify the ambient air. Equipment for a heat pump water heater consists of a refrigerating compressor, cooling surface, expansion valve, condenser water storage tank and connecting piping. Either a forced or natural circulation type cooling surface can be employed. A vertical or horizontal condenser water storage tank can be used as desired.

Design Features

There are quite a few possible variations to the four basic designs — precooling coils, pre-heating coils, or humidifying may be incorporated in the design. Two condensers or two evaporators may be used in the first two designs, to prevent reversing the fluid flow, or to heat one section of the space and cool another section simultaneously. It is possible that the use of four 3-way valves or four check valves and four 2-way valves instead of the eight 2-way valves shown in the diagrams would result in a saving in equipment and installation cost but, on the other hand, may not always be the most satisfactory. Since cold fluid is on one side of the valve and hot on the other, the 2-way valves seem to offer less chance of leakage across the valve seat and also usually offer less resistance to fluid flow.

The usual precautions of a conventional refrigeration system must be followed in a heat pump design. It is advisable to use oil traps ahead of the compressor, and to size and install the

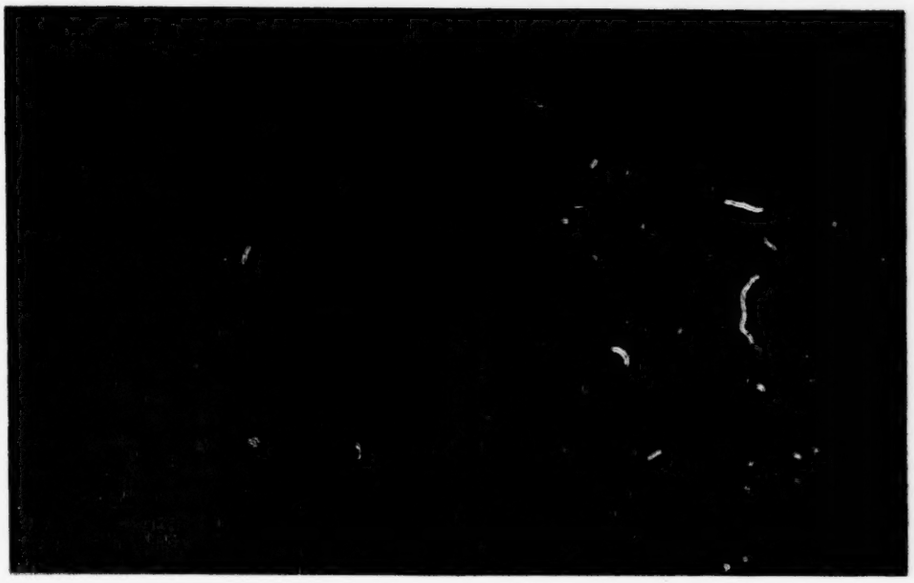


Fig. 6. Layout of heat pump for heating domestic hot water with delivery hot water temperatures of 140-150F.

refrigerant piping to prevent trapping oil in the system.

In the first two designs the surfaces have a dual role of acting as an evaporator and a condenser, so that special care must be exercised that they are self-draining to prevent trapping of refrigerant and oil. In these two designs the expansion valve is exposed to high temperature-high pressure refrigerant which requires special consideration.

In all the designs it is important to have the equipment selected, arranged and fabricated to conform with the six basic qualities of:

1. High reliability.
2. Economical performance. The unit must be highly efficient and subject to inherently low maintenance cost.
3. Reasonable first cost.
4. Compactness. All working parts must be readily accessible for repair and maintenance.
5. Neat appearance.
6. Low noise level and minimum vibration.

In all the designs, it is quite important to have an efficient, quiet, trouble-free compressor (preferably hermetically sealed type) operating at reasonable speeds and equipped with capacity modulation features to obtain as wide an operating range as possible.

Refrigerant and water piping should be so designed that seasonable changeover is easily accomplished. Refrigerant valves and check valves for direction of refrigerant flow should be trouble-free, positive acting, and have a minimum pressure drop. This changeover can be automatic or manual, but usually automatic is preferred.

The heat pump is applicable to most any type of distribution system from the unit to the conditioned space. An air system circulating warm or cold air by means of ducts, or a water system circulating warm or cold water by means of pipes to one or more units located in the conditioned space, have been successfully installed and used.

Controls

Control of the heat pump system is important to maintain the desired temperature and humidity during both the heating and cooling cycle. If air is the heat source, provisions should be made for the defrosting of the outdoor coil when required. Controls should be reliable, sturdy, trouble-free and positive. There are many methods of control ranging from manual to completely automatic operation. Either an electric, a pneumatic or a combination electric-pneumatic control is applicable to a heat pump. Obviously the minimum amount of control equipment should be used to achieve the operation and performance desired. Sufficient safety controls should be incorporated, however, to prevent excessive head pressure and objectionable low refrigerant temperatures. Usually in designs using outdoor air as the heat source, additional controls must be incorporated because, as the outdoor temperature rises, the available capacity increases and the heat requirements decrease.

Fig. 7. Comparative heating cost of electric resistance heating, heat pump, oil, coal and gas (on basis of 60% burning efficiency for oil, coal and gas), and for various heat pump coefficients of performance (C_p).

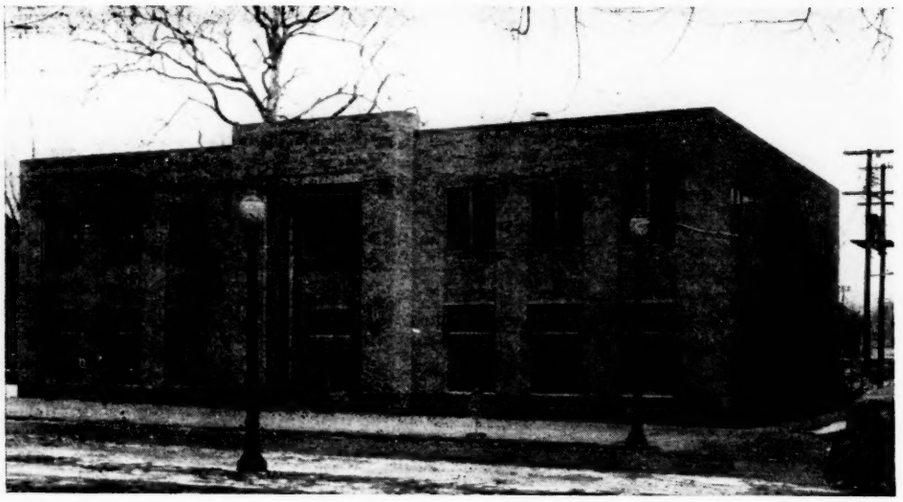


Fig. 8. Utility company office building at Coshocton, Ohio, heated and cooled by a water-to-water heat pump as shown in Fig. 4.

The advantage of one control arrangement design over another depends to a large extent on the type of building and location of the system.

Why a Heat Pump

Electric resistance heating long has been considered the ideal from the standpoint of the heating customer because the advantages it offers over other fuels of (1) elimination of dirt, soot and fumes; (2) elimination of firing and other furnace labor; (3) elimination of fuel and combustion products from the premises; (4) accurate and reliable room temperature regulation; (5) minimum heat losses in the basements and other parts of the structure where heating is not required, and (6) minimum maintenance and repair.

Unfortunately, however, even with the many inherent advantages of electric resistance heating there are very few areas in the United States where it is being used or can be justified at the present time when the relative cost of delivered electric energy is compared with the corresponding cost of coal, oil, gas or other fuel. Electric energy is under a great handicap because of the cost of the distribution system necessary to supply energy on a one to one conversion basis. Steam generated electric energy has a further disadvantage when attempting to compete in the heating field with directly released heat energy because of the low conversion efficiency of the power plant. The most modern steam generating plant has a conversion efficiency of only 34%. If an allowance of 15% is made for transmission and distribution losses, this is reduced to an overall conversion efficiency of 29% of the original heat content of the fuel. This compares with a probable percentage recovery of 50 to 80% in direct fired heating installations.

The heat pump offers a way of eliminating the handicap that electric energy is under when it attempts to enter the residential or commercial heating field on a straight energy conversion basis by furnishing 3 to 5 kwh of equivalent heat for each kwh of energy used. This makes possible and wholly practical the supplying of electrical heating service to residential and commercial establishments without sacrificing any of the advantages of electric resistance heating.

Potentials of Heat Pump

The development of the heat pump is important to the electric utilities, to the manufacturers of refrigerating equipment, and to the customer.

As a potential load for the electric utilities, the heat pump offers a desirable and profitable residential load exceeding many times the present residential electric energy consumption of all the other electric appliances combined. There was an estimated total of 34 million residential heating installations in the United States for the year 1940, consuming approximately 2,382 trillion Btu of useful heat. Thus useful heat, when converted to equivalent electrical energy units, represents an annual energy consumption of 697 billion kwh. This means that a 4% saturation of the heating customers with heat pumps would approximately equal the total residential electric consumption for 1944 (estimated at 29.7 billion kwh).

The heat pump offers to the manufacturer of refrigeration equipment the possibility of manufacturing a self-contained unit usable the year round for both heating and cooling. This type design permits quantity production and improved production methods which will result in a more attractive selling price to the consumer. The growth that can be accomplished in the air conditioning industry by means of the heat pump can be readily appreciated when it is realized that during the first 11 months of 1935 only slightly more than 40,000 tons of compressor capacity were installed in the entire country for air conditioning purposes. Practically none of the capacity was used for residences.

The heat pump offers the customer a self-contained year round air conditioning system providing cooling, dehumidification, air circulation, air cleaning, heating, and humidification. The heat pump maintains the many advantages of electric resistance heating stated and possesses the additional

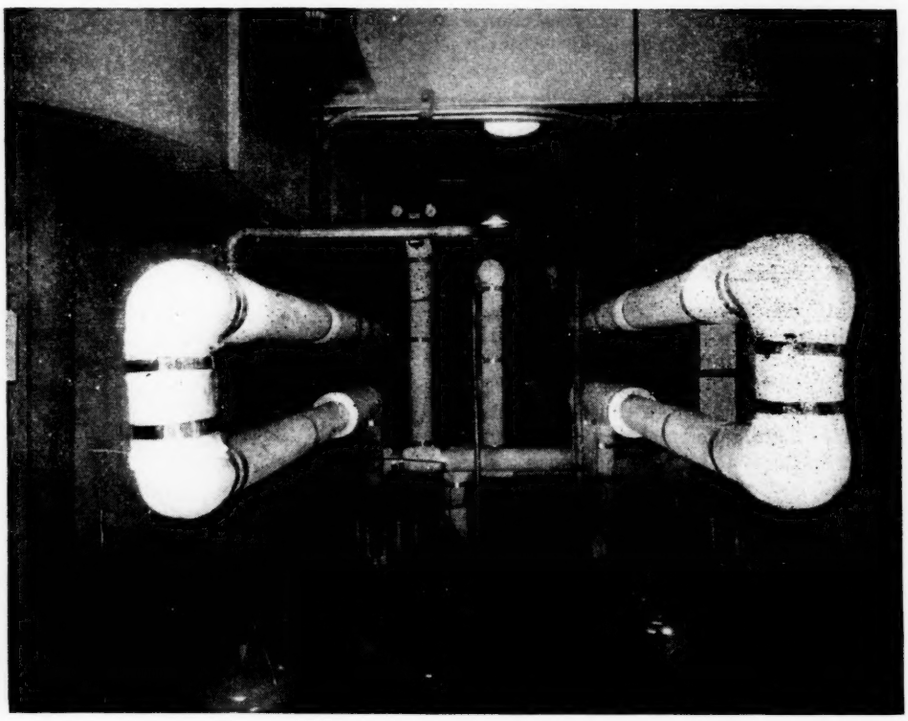


Fig. 9. Heat pump equipment room in basement of Coshocton building, showing 10 and 15 hp. compressor; evaporator and control in background.

advantages over a conventional heating and cooling system of:

1. Compactness — The combination of heating and cooling in one system results in a minimum use of space.
2. Flexibility in arrangement—The location of the equipment is not fixed. It can be on the roof, in the basement and, in fact, in almost any part of the building.
3. Concentration of utility service—By combining heating and cooling requirements into an all year round system it is possible to arrange the building to operate on a single electric utility service. Thus the need for gas or oil is automatically eliminated.

When air is used as the source of heat, defrosting of the outside air coil is usually necessary when the outdoor temperature falls below 35F. Seven methods of defrosting the outdoor coil are:

1. Reversing the refrigeration flow.
2. Spraying with water at 50F or above.
3. Dehumidifying the air with solid absorbents before it passes over the coil.
4. Dehumidifying with liquid absorbents.
5. Use of electric strip heaters.
6. Use of two coils, one being in use while other is defrosting.
7. Storage of heat in tanks using off-peak energy from the heat pump.

In the off-peak storage system it is possible, when on the heating cycle, to store hot water during mild weather for use when colder outdoor temperatures are encountered and when on the cooling cycle to store cold water during mild weather for use when warmer weather is experienced. It is also possible to store either hot or cold water, depending on the cycle of operation, during the night or off-peak periods for use during the periods when the heating demand is greatest. It is also possible in the off-peak system to defrost the outdoor coil without interrupting the delivery of the heat to the conditioned space.

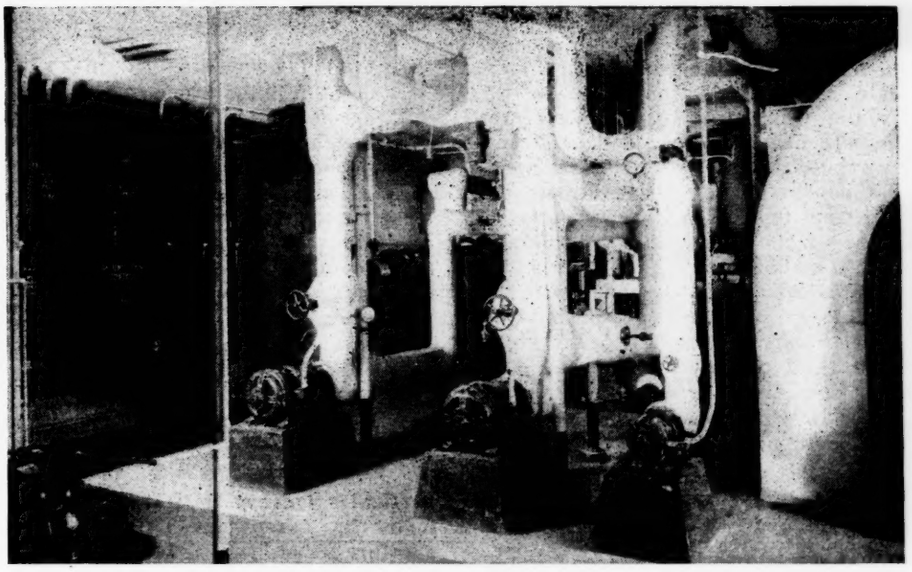


Fig. 11. Air conditioning equipment on the fourth floor of Portsmouth building, and which operates at an average coefficient of performance of 3.0.

Heat Sources for Heat Pump

An ideal heat source is one which is abundant and inexpensive with an average temperature of 40 to 80F the year round. Outdoor air, ground and water from wells, lakes and rivers may meet the requirement. Discharged water from manufacturing processes or any other liquid where the temperature is too low for direct utilization may qualify providing it is chemically suited and does not require special metals or extensive treatment.

Outdoor air offers possibilities as a heat source in locations where the minimum temperatures are not too low or of too long duration. In many locations it may be practical to offset the resulting loss of capacity caused by extremely low and wide fluctuations in daily outdoor temperature by the use of an auxiliary heat source such as the sensible heat of water, latent heat of freezing, and electric resistance or fuel fired heaters.

Well water temperature is fairly constant year round and would be an ideal universal source if it were chemically satisfactory and if it could be obtained at a low cost. Unfortunately there is always the uncertainty of finding water at any given location and the cost of drilling and the maintenance involved in the use of water further detract from its use in many places. Also, the disposal of the water after its use may constitute a serious practical problem. In many locations it may be found that due to the cost of drilling a well or the cost of obtaining suitable water, one of the other heat sources will be more practical.

Ground offers possibilities as a heat source for heat pump systems, but to date very little data are available on the conductivity of heat to and from the soil. Considerable more data are necessary before this heat source can be evaluated properly. Several universities, manufacturers, and national engineering societies are investigating the ground as a heat source and plan to make actual installations to determine the transfer coefficients for different types of soil and at the same time obtain some knowledge on the cost and performance of the systems. Considerable data are available on the heat loss from oil piping, electrical conduits and the like which have been buried in the ground. Such data may be helpful in obtaining conductivity

Fig. 10. Office building at Portsmouth, Ohio, with an air-to-liquid heat pump as shown in Fig. 3.

coefficients for heat pump systems when the pipes in the ground are used as a condenser. However, very little data are available on pipes in the ground when used on an evaporator to take heat from the soil. Information published to date indicates that the conductivity of heat directly to and from the soil is quite variable because of the wide range of climatic conditions and geological formations found throughout the country. The heat loss conductivity coefficients in Btu per hr per F per sq ft per ft which have been compiled for buried pipe lines indicate a range from 0.14 for dry white sand to 2.62 for a very wet soil.

Comparative Heating Costs

A comparison of the heating cost of a heat pump with a conventional heating system is rather difficult because the heating cost of a conventional system is quite variable. It depends upon the type of fuel used, type of equipment employed and the care with which it is operated. Fig. 7 shows the comparative heating cost in dollars per million Btu of electric resistance heaters, the heat pump, and of coal, oil and gas systems. Fig. 7 shows that a heating cost of $1.25 per million Btu will be obtained if:

1. Electric energy using resistance heating is 0.43¢ per kwh
2. Electric energy using heat pump (coefficient of performance = 3) is 1.28¢ per kwh
3. Electric energy using heat pump (Cp = 4) is 1.72¢ per kwh
4. Electric energy using heat pump (Cp = 5) is 2.14¢ per kwh
5. Electric energy using heat pump (Cp = 6) is 2.57¢ per kwh
6. Gas (550 Btu per cu ft) with 60% burning efficiency is 41.2¢ per M cu ft
7. Gas (900 Btu per cu ft) with 60% burning efficiency is 67.8¢ per M cu ft
8. Oil (140,000 Btu per gal) with 60% burning efficiency is 10.5¢ per gal
9. Coal (12,000 Btu per lb) with 60% burning efficiency is $18.05 per ton

A heating cost of 0.75 dollars per million Btu will be obtained if:

1. Electric energy using resistance heating is 0.26¢ per kwh
2. Electric energy using heat pump (Cp = 3) is 0.77¢ per kwh
3. Electric energy using heat pump (Cp = 4) is 1.03¢ per kwh
4. Electric energy using heat pump (Cp = 5) is 1.28¢ per kwh
5. Electric energy using heat pump (Cp = 6) is 1.54¢ per kwh
6. Gas (550 Btu per cu ft) with 60% burning efficiency is 24.7¢ per M cu ft
7. Gas (900 Btu per cu ft) with 60% burning efficiency is 40.5¢ per M cu ft
8. Oil (140,000 Btu per gal) with 60% burning efficiency is 6.3¢ per gal
9. Coal (12,000 Btu per lb) with 60% burning efficiency is $10.80 per ton

Although the tabulation given by Fig. 7 does not consider all the phases of the competitive problems it does clearly show that the heat pump with all its advantages can be brought on a full competitive level with the more conventional, less flexible heating systems.

Heat Pump Installations

There are several hundred heat pump installations throughout the United States using air or water as the heat source depending upon the particular locality in which they are installed. These heat pump systems are furnishing heating, cooling, filtering, dehumidifying and all the other functions of year round air conditioning systems for office buildings, commercial establishments and homes. Two heat pump systems of modern design which have been operating satisfactorily since their installation in 1940 are located in the office building of The Ohio Power at Portsmouth and Coshocton, Ohio. The Coshocton office building has two stories and basement and is 88 ft long by 55 ft wide by 35 ft high. The Portsmouth building is 104 ft long by 45 ft wide by 45½ ft high. Both buildings are brick construction with 1 in. of insulation on all external walls and roof. All windows are fixed and have double glazing. The heat pump equipment is located in the basement of the Coshocton building and on the fourth floor of the Portsmouth building.

The Coshocton system uses a water to water design, shown by Fig. 4. The blow through type conditioner consists of two plenums; a heating coil and humidifier in one plenum and a well water precooling coil and a mechanical refrigeration coil in the other. The hot and cold plenum connect to the various zones throughout the building by means of sheet metal ducts. The zone thermostats maintain the desired temperature by controlling their respective volume dampers located in the ducts connecting to each of the two plenums. A 10 and 15 hp, freon F-12 refrigerating compressor furnishes the heating and cooling for the Coshocton building.

Fig. 8 is an external front view of the Coshocton building and Fig. 9 is a view of the equipment showing the 10 and 15 hp compressor in the foreground and the evaporator and control in the background. The conditioner containing the fans, filters and conditioner coils is at the right in Fig. 9. The average coefficient of performance obtained for the system by actual performance test is 4.0 when deepwell pump, circulating pump and auxiliaries are not included. An average coefficient of performance of 3.3 is obtained when compressors and all the auxiliaries are included. In this system the deepwell pump has an electrical demand of 3.34 kw, the compressors 10.2 and 15.65, and the circulating pump 1.84 kw.

The Portsmouth system uses an air to liquid design shown by Fig. 3. The same type of conditioner and zoning arrangement used at Coshocton is used at Portsmouth. Two 25 hp, freon F-12 refrigerating compressors furnish the heating and cooling requirements for the Portsmouth building.

The outdoor unit for the Portsmouth system consists of a coil, an automatic intake and exhaust damper, a circulating fan and spray nozzle. During normal operation outdoor air is circulated over the coil. When the outdoor temperature falls below a predetermined temperature during the heating cycles or rises above a predetermined temperature during the cooling cycle, city water can be supplied to the spray nozzles to act as an auxiliary heat source or to give evaporative cooling.

Fig. 10 shows the external front view of the Portsmouth system and Fig. 11 shows the heat pump equipment room. The control board is to the right in Fig. 11, the circulating pump in the foreground and the 2-25 hp compressor, condenser and liquid cooler in the background. A portion of the outdoor air fan is shown to the right in the figure. The average coefficient of performance for the Portsmouth system by actual test is 3.5 when compressor input alone is considered and 3.0 when compressor, pump, and all auxiliaries are included.

ECONOMIC POSSIBILITIES

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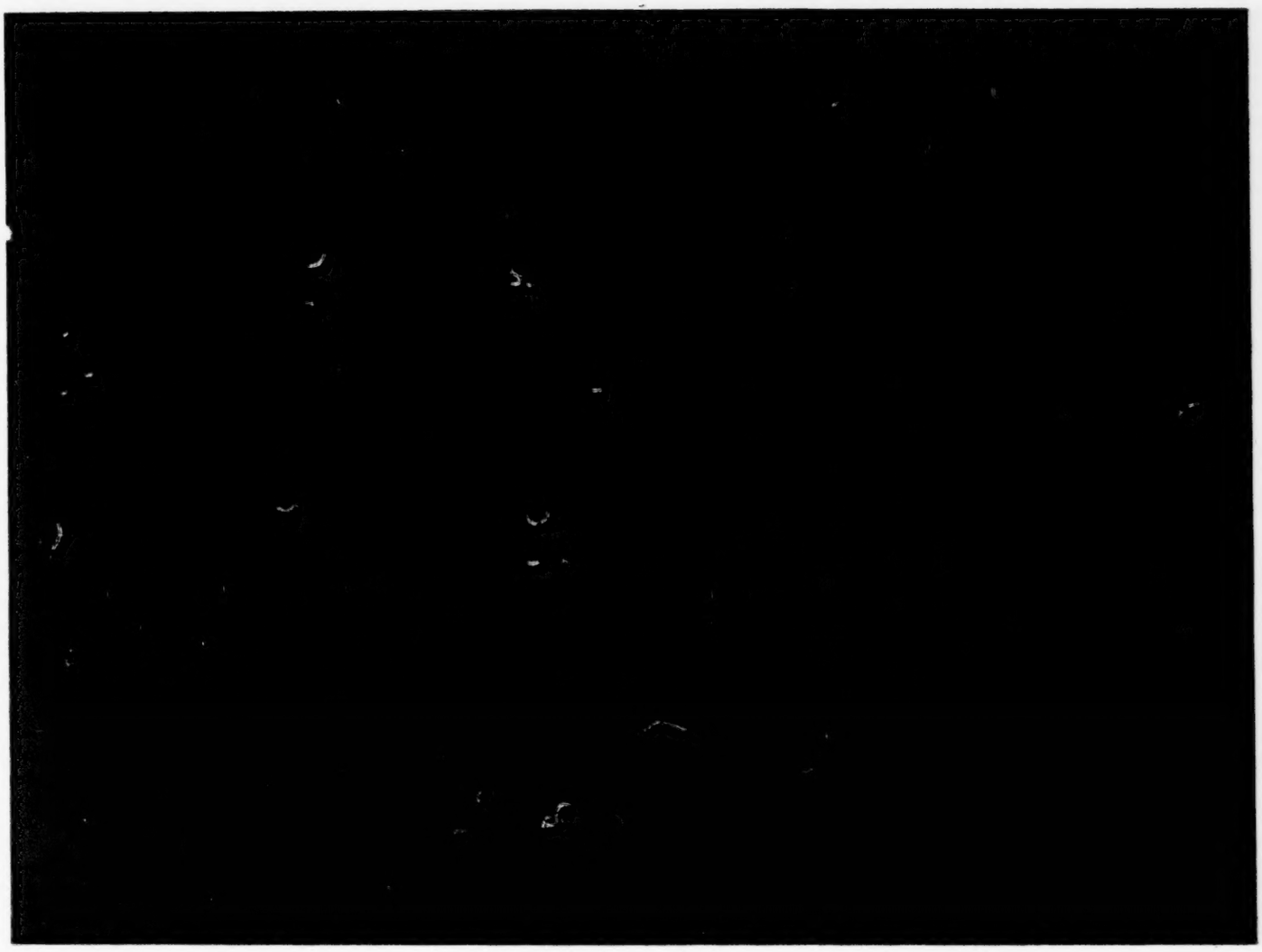
DURING the last few years there have been many developments which have changed the outlook for the heat pump. From the mechanical side are improved refrigeration compressors. Higher operating speeds make it possible to build machines which are much smaller than previous units. There have been developed hermetically sealed compressor units of a capacity readily adaptable to the size of heat pump which will fit the five and



six room house, and for which there will be the largest sale. Secondly, there have been developed improved heat transfer surfaces and designs which will considerably reduce the weight and size of the heat exchangers which form so important a part of the heat pump. These two improvements, therefore, now make it possible to build a physical unit which will fit easily into the home.

Sources of Heat†

Air. One of the universal sources of heat, and perhaps the most common for heat pump usage, is the surrounding air. Because of its convenient and general availability and ease of movement it appears to be at first thought a general answer to the problem. However, heat pump efficiency drops off rapidly as the source temperature decreases. Since heating requirements increase as the outside temperature drops, the air-to-air unit requires increased size in order to make up for this decrease in efficiency as the output requirements increase. Where heating is the controlling load on the heat pump, and where low outside air temperatures are encountered, the air-to-air unit will require a larger horsepower capacity unit than the water-to-air unit. Where the cooling load imposes the maximum load, the air-to-air unit may not be penalized in this respect.

The air-to-air unit is basically larger in size than the water-to-air unit because the air heat exchanger is larger than the liquid heat exchanger and provision must be made to bring the outside air to, through, and from the unit. Because the temperature drop in the air from which heat is abstracted must be kept low, large volumes of air must be handled. This introduces noise problems and unless the heating unit is conveniently located to an outside air source will require a considerable amount of power.

Practical considerations would appear to limit the air-to-air unit to temperatures of from 10 to 20F. There are, however, many possible modifications of current heat pump cycles which may result in improved performance and capacity for lower temperatures.

Ground Water. Ground water as a general source of heat for heat pump operation is not practical unless some means is used to return the water to the underground reservoir. Any general widespread usage of ground water resources without returning them would in most localities result in depletion of these resources and would not be in the public interest. There are specific areas, however, where there are sufficient ground water resources to permit pumping and discarding them. This is particularly true in coastal regions. It is also possible to utilize dual wells so that water can be pumped from one and returned to another. The practicality of this recycling will depend upon local conditions, well spacing, etc.

The general advantage of utilizing water as the source of heat is the fact that it will be at essentially the same temperature during the entire heating season, which results in a more compact unit. By having a uniform source temperature, it is possible to build a unit which can be designed to operate at capacity.

Earth. The most universal source of heat which is available is the earth. A cylinder of earth 30 feet in diameter and 100 feet deep will give up 28,000,000 Btu if its temperature is lowered only 10F. If we assume a 100-foot depth and a heat pump unit delivery of 50,000 Btu per hour operating an average of 20% of the time over a five-

†Material under this heading has been abstracted from the author's report "A Survey of Heat Pump Sources" referred to in the footnote to Table 2.

month heating season, the drop of 10F in a 30-foot diameter cylinder would approximately supply the heat needed for the entire heating season if no heat flowed into the region. There is no question as to the capacity of the earth to supply the heat; the problem is to obtain this heat.

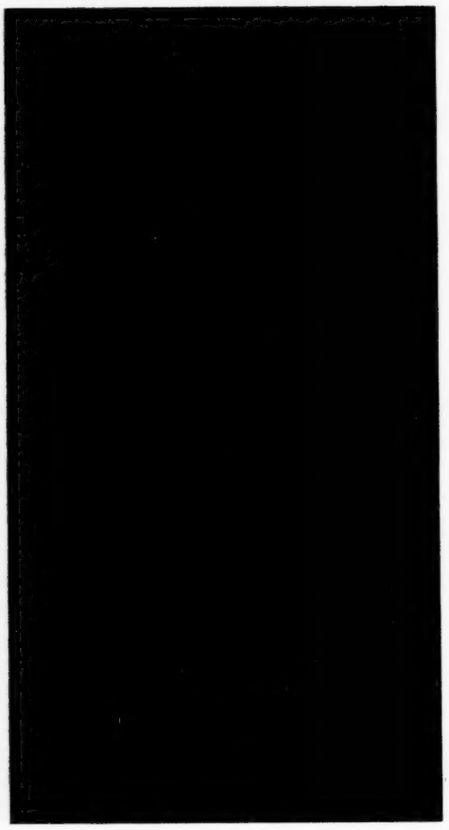
Other Sources. There are many other possible sources of heat, most of which depend upon particular local conditions and have only limited application. However, where they can be utilized, the heat pump installation may be simplified and in many cases where higher temperature sources are available, higher coefficients of performance may be obtained. In Switzerland lakes and rivers have been used as the source of heat for large installations. Other installations have used individual water supplies, waste waters, and heat from industrial processes.

New Building Design Possible

The heat pump offers to the architect many opportunities in creative design of buildings. If the architect can adapt his ideas to the full extent of the heat pump possibilities, a type of home which will be entirely different in construction can be evolved. Particularly for the housewife it will have many advantages not currently attainable. The fact that there is no combustion at all in the home offers home construction without chimneys, vents, etc. The heat pump, particularly water-to-air units and probably future air-to-air units, will be of such size that they can be mounted satisfactorily anywhere within the house. Over a period of time, research will undoubtedly find a way to reduce the size of the air-to-air unit and it is entirely possible that it can be mounted in the attic where a very convenient source of air is available.

This elimination of a basement requirement for proper heating and cooling equipment will be a definite advantage in many areas. Where basements are not particularly desired by the occupants, it will permit a much lower construction cost because of elimination of the basement; the use of much smaller space on the first floor; the possible utilization of currently waste space in the attic.

By providing year-around comfort and by making provision for continued air circulation in the home, it is possible to go into entirely different types of construction eliminating, if desired, all window openings through which a large part of the dust gets into the home. By eliminating combustion, soot and smoke, it is conceivable that dusting can be entirely eliminated from the present home routine. To realize the full possibility of the heat pump will require some readjustments in our concepts of heating and cooling and will necessarily call for a readjustment in our ideas of home comforts and construction. The heat pump offers a challenge to the architect to provide a design that will capitalize on its many advantages.



Heat Pump Performance

It appears that heat pump performance of better than 4 to 1 on the heating cycle under normal operating conditions as would be encountered in most of the country is entirely possible. It is likewise probable through technological developments to boost this considerably above 5 to 1 if it should prove to be necessary. The heat pump, therefore, as a device has in reserve many possibilities which have not to date been explored. This is decidely in its favor.

If, however, the general current designs are maintained, there is every indication that these units, when available with hermetically sealed compressors and other new developments, should operate as trouble free as the ordinary household refrigerator. It should be noted that the current household refrigerator which is a complicated piece of equipment, in principle has many additional features over those of the current heat pump. Even though technological additions are made to the heat pump, there is every indication that the heat pump should still continue to operate trouble free and as satisfactorily as the household refrigerator. With coefficiency performance between 4 and 5, and investment costs kept only slightly more than an adequate heating plant, the heat pump appears to offer very attractive possibilities to those who want combined heating and cooling.

The heat pump also appears to have possibilities with regard to its application to the radiant heating field. Because of the generally low temperatures which are required in radiant heating, the heat pump should be able to fill this requirement. If water is used as a source of heat, two liquid heat exchangers could then be utilized, resulting in a most compact and economical set-up, particularly when used with a hermetically sealed compressor. This means that for adequately taking care of a moderately sized home, a unit which could be set in a closet or utility room would be practical.

As with many new devices, the heat pump offers possibilities of developing new outlets for automatic heat and year-around comfort. One of these important and rapidly increasing markets is in rural electrification areas. In these areas, the heat pump offers a type of automatic heat not otherwise available. Since cooling can also be provided at little extra cost, it would appear to be a particularly adaptable means of rural home comfort.

Non-User Interests

The development of the heat pump is of interest not only to manufacturers of heating and ventilating equipment, but also to coal producers and the electric utilities. To heating and ventilating equipment manufacturers, it offers the possibility of developing not only equipment for heating and cooling homes, but also eventually developing multi-purpose heating and cooling equipment for home and commercial uses. There are likewise many industrial applications of the heat pump which have development possibilities.

Electric utilities are vitally interested in the heat pump. It offers for the first time the means of realizing the all-electric home. While the heat pump would mean a large increase in the load for utilities, it would also create numerous problems for them. To evaluate the present status of heat pump development and to predict its future possibilities, the Southeastern Electric Exchange, the association of electric utilities in the southeastern part of the United States from Virginia to Louisiana, have employed the Southern Research Institute to undertake a general survey of the heat pump.

Research Program

As a part of this program, the Southern Research Institute has made a survey of manufacturers of heating and ventilating equipment to find out whether anyone is working on the heat pump. As a result of this survey, over 130 replies have been received and out of this group, 13 manufacturers reported that they are at present carrying on research work connected with some phase of heat pump development. An additional 10 manufacturers in-

dicated that they had intentions of doing heat pump research in the near future. In addition to this group, there are several companies not covered by the replies who are also doing development work in connection with the heat pump. This indicates that a tremendous interest has developed recently in this field and it points to use of the heat pump as a large scale commercial device.

As a part of its general program for the Southeastern Electric Exchange, the Southern Research Institute plans to make preliminary investigations of such questions as sources of heat, off peak storage, current and voltage requirements of heat pump installations, survey of installations which have been made, temperature and design data as it influences the heat pump, theoretical cycles, etc., as well as testing such commercial units as are available on the market.

Since this program as undertaken by the Southeastern Electric Exchange is a general contribution to the heat pump cause, the information will be made available through special reports or the technical press. A bibliography on the heat pump has been compiled and is available to those interested.

Present activities today both on the part of manufacturers and electric utilities point to an orderly and rational development of the heat pump. The interest and activity which the utilities are taking in this development will play a very important part in insuring the manufacturers of cooperation in the development of the heat pump and in its application.

THE MARVAIR UNIT

M. M. SMITH
Vice-President, Muncie Gear Works, Inc., Muncie, Indiana

AN automatic air conditioning plant, termed Marvair giving year-around comfort has been developed by the Muncie Gear Works. This unit takes heat from the earth in winter and puts it in the home; in summer it takes heat from the home and puts it in the earth. Marvair represents the results of several years' research and development work as well as home testing.

It employs the simplest type refrigeration circuit, the basic elements of which include expansion valve, water radiator, air radiator, and compressor. To this basic circuit must be added a water system connected to the water radiator to furnish heat in the winter and to dispose of heat in the summer. There must likewise be an air circuit connected to the air radiator to carry air to or from the space to be conditioned.

The general system used by Marvair is shown in Fig. 12. It consists of a fan in the unit marked "Fan Housing," which furnishes the propelling force for getting the air to and from the space to be conditioned. The equipment shown in Fig. 1 is in the housing marked Marvair. A pump, as indicated on the water side, circulates water to the water radiator. This water system will be discussed in greater detail.

As was indicated, Marvair employs a simple heat pump cycle. It furnishes heating or cooling as required by simply interchanging the functions of the water and air radiators. This is accomplished physically by using a very simple piping circuit and four solenoid valves. The change from heating to cooling requires only the interchange in operation of these two pairs of solenoid valves.

Fig. 13 shows the piping diagram when operating on the heating cycle. For this condition valves 1 and 2 are held open and valves 3 and 4 are kept closed. When operating on the heating cycle the compressor discharges the hot gas through open solenoid valve 2 to the air radiator. The air in flowing over this air radiator absorbs heat. In giving up this heat the refrigerant gas is condensed and returns through the check valve, as indicated, to the receiver. As more heat is required, the expansion valve admits this liquid to the water radiator. This water radiator is in effect the receiver. The refrigerant is vaporized in the water radiator and is returned through solenoid valve 1 which is open to the compressor. This represents the circuit and method of operation of Marvair unit on the heating cycle.

The flow circuit for the cooling cycle is shown in Fig. 14. When operating on the cooling cycle, solenoid valves 3 and 4 are open and valves 1 and 2 are closed. For this condition, it can be seen that the compressor discharges the hot gas through valve 4 to the water radiator where the refrigerant is condensed and delivered through the check valve to the receiver. As more cooling is required, the expansion valve admits the liquid to the air radiator. As the refrigerant is evaporated in the air radiator, it absorbs

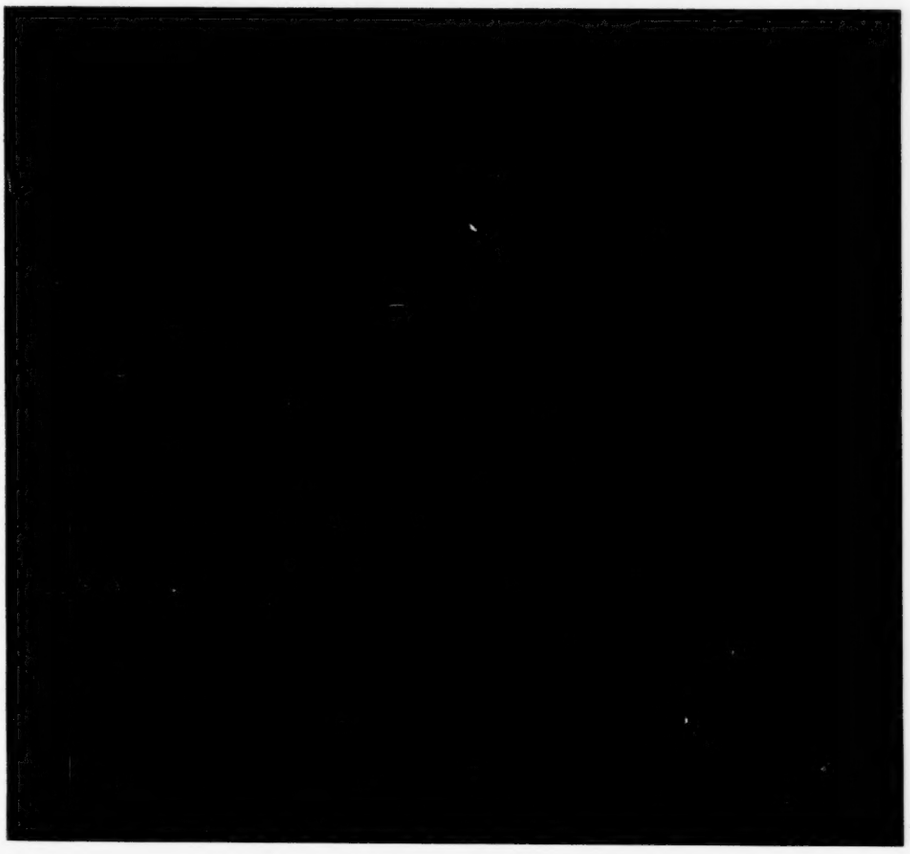


Table 3—Test Installation Data

Pump discharge pressure, lb per sq in.	30
Pressure drop across heat exchanger, lb per sq in.	2
Pressure drop across water meter, lb per sq in.	4
Pressure drop in pipe system, lb per sq in.	19
Electric consumption of motor, kw	0.39
Water flow, gph	626

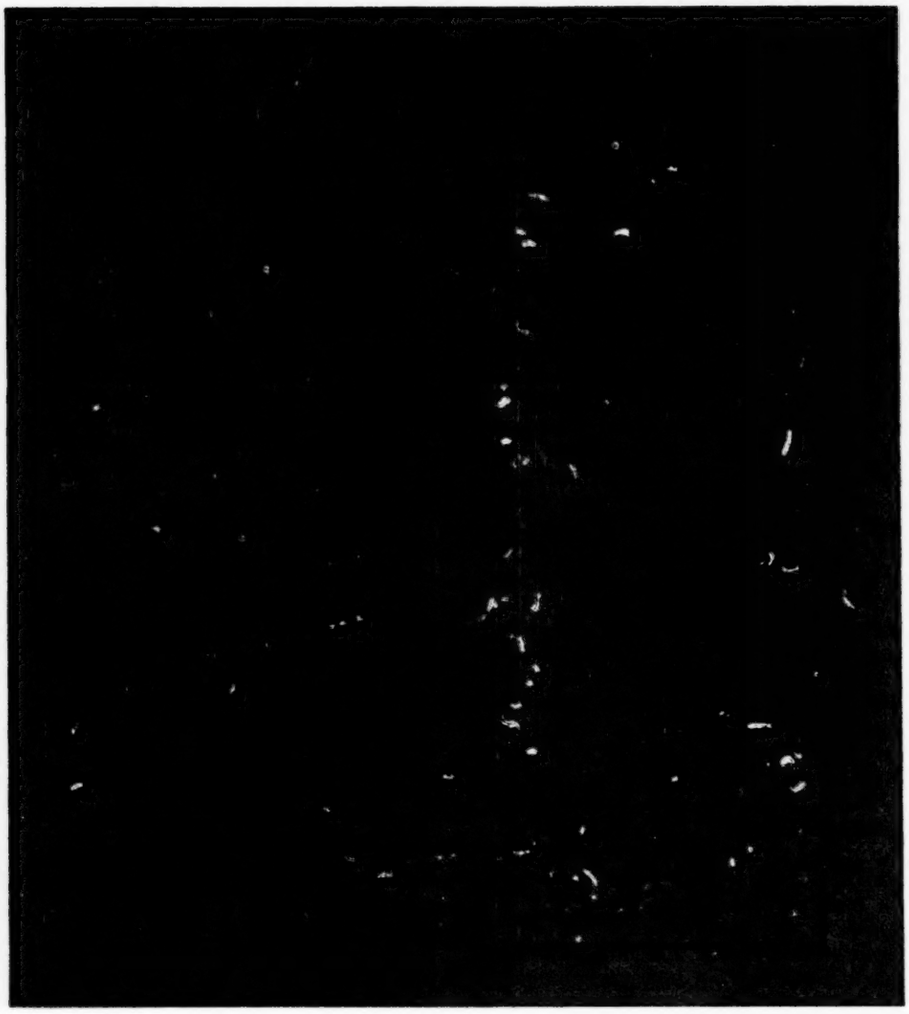


Fig. 13. (Upper). Piping diagram for Marvair when operating on the heating cycle. Open and closed solenoid valves are shown. Fig. 14 (Lower) Piping diagram for the cooling cycle with the position of valves indicated.

provided in case some failure of water supply would tend to result in freezing in the evaporator. The automatic controller utilizes the mercury type switch to prevent switching troubles. The controller is actuated as indicated by a thermostat and the controller, by its position, operates contacts which in turn actuate the solenoid valves in pairs as indicated.

Everywhere in this circuit standard elements are utilized to insure satisfactory operation and to eliminate sources of trouble wherever possible. As can be seen from this diagram, this electrical control circuit is exceedingly simple when it is considered that it provides automatic heating or cooling, protects the compressor motor against excessive currents and excessive or deficient pressures, and likewise protects the evaporator against freezing due to water failures or other cause.

The most important part of any heating system is the source of heat. In a furnace, it is coal, oil, or gas. In a heat pump, the difference between the electrical energy supplied and the heat output must be supplied from some source. Marvair utilizes the heat stored in the earth as its heat source.

Fig. 12, in addition to the general scheme, shows the method employed by Marvair for getting heat from the earth. The water system as indicated is very simple, consisting of a closed pipe circuit and a water pump. This set-up uses a closed system which has many advantages. Since no water is pumped from the ground, there is no effect on the water level and no problem of disposing of water. The closed system requires only an amount of heat. This heat is taken from the air. The refrigerant gas is then returned through valve 3 to the compressor.

It can be seen from the above description that the change from heating to cooling is accomplished by a simple interchange of the evaporator and condenser functions through the operation of the four solenoid valves and two check valves. When heating, the air radiator is the condenser and the water radiator is the evaporator. When cooling, the water radiator is the condenser and the air radiator is the evaporator. This general method of accomplishing the changeover from heating to cooling is designed to give trouble-free service and to keep the initial cost to a minimum. The corresponding control circuit is likewise exceedingly simple and as trouble-free as it is possible to make such a control.

Fig. 15 shows the basic electrical control circuit. The general circuit includes line disconnect and start-stop switch connected ahead of main contactor. In order to protect the compressor motor, over-current trips are provided as well as over-pressure and under-pressure trips. To protect the water radiator and water pump motor, a water temperature cut-out switch is

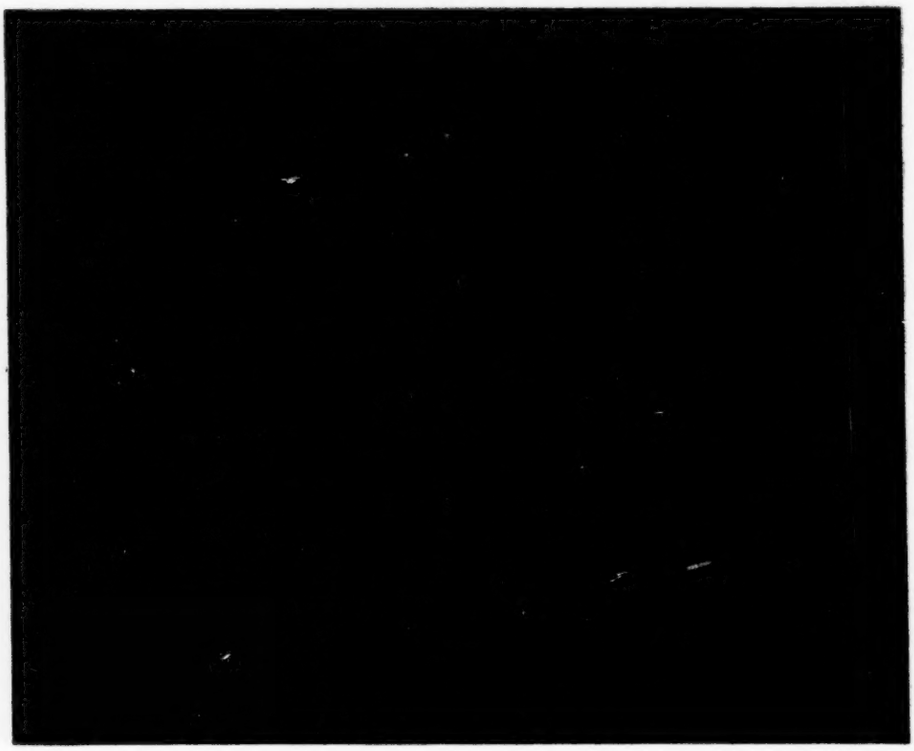


Fig. 15. Basic electrical control circuit for Marvair that provides for automatic heating or cooling.

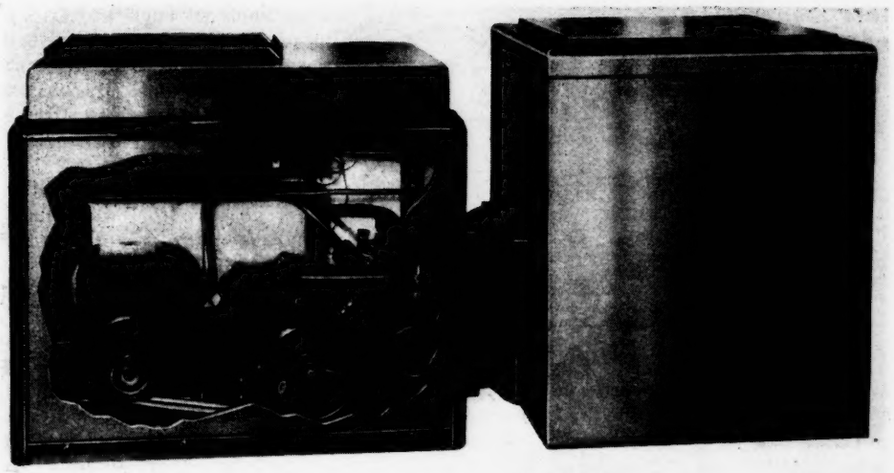


Fig. 16. Marvair unit for automatic air conditioning with part cut away to show the compressor.

Table 4 — Marvair Specifications

(Subject to Change)

Motor Size— HP	Width In.	Length In.	Height In.	Weight Lb.	Btu Heating Capacity— @ 50F Water	Btu Cooling Capacity— @ 75F Water
3	28	40	34	845	60,000	32,000
5	28	40	34	1075	100,000	54,000

Table 5 — Fan Specifications

Marvair Size— HP	Motor Size— HP	Width In.	Length In.	Height In.	Weight Lb.	CFM	Static Pressure— In.
3	1/3	21 3/4	27	22	87	1500	3/4
5	1/2	21 3/4	27	22	87	2000	3/4

power to overcome friction in the pipes, since no fluid is raised from the well bore. As this is a closed system, an inhibitor is used and corrosion completely eliminated. Further, since the fluid in the system can be treated, heat exchanger inefficiencies as a result of scaling, corrosion, etc., are likewise completely eliminated. All of these factors lead to low operating cost and long trouble-free life.

Table 3 lists some test data taken on a system of this type. The amount of water which must be circulated depends in part on the earth temperature. At the most extreme condition to be encountered in most of this country, 600 gph would satisfactorily take care of a 3-hp unit and in many parts of the country would take care of a 5-hp unit.

Equipment as now supplied is furnished in two units. The first includes the fan only, and the second the refrigerating equipment (marked Marvair in Fig. 12). This is done in order to provide ease of installation. Each of these units is very compact, as can be seen from Tables 4 and 5, which give specification data on the Marvair and fan units. Utilization of the earth as a source of heat insures maximum performance under conditions requiring the maximum heat quantities. If higher water temperatures are available, the capacities of the unit will be slightly higher.

In the application of the Marvair a new concept of use of heat quantities is involved because of radical departure from current practices. Under such circumstances it was felt that the rating of this unit could best be accomplished by adapting the performance figures to current methods of calculating heating loads. Because this unit does not require air for combustion and provides a simple means of air circulation, a given heating capacity in Marvair can supply a much greater capacity than would otherwise be indicated by standard methods of calculation.

This revolutionary air conditioning plant is ideal for homes, offices, and small commercial buildings, and it provides perfect comfort in June or January through the operation of one small, compact, electrically operated installation. Its advantages in cleanliness, safety, and convenience are readily apparent. It eliminates grime and drudgeries which have been associated with heating through the ages and adds pleasant temperature for summer living to warmth for winter.

Marvair is now a reality, after years of effort on the part of a large group of heat exchange experts, engineers and skilled mechanical workers. The manufacturers are assured of the fact that an eagerly receptive market awaits this new product, which places science's greatest advance in year-around air conditioning within the reach of many.

THE AIRTOPIA UNIT

W. L. HOLLADAY

Chief Engineer, Airtopia Distributors, Inc., Los Angeles, Calif.

AIRTOPIA is a reversible cycle air conditioning unit, using air for heat pickup and heat disposal. It performs the six functions expected in comfort conditioning: Cooling, heating, circulating and filtering air, as well as humidification on the heating cycle and dehumidification on the cooling cycle. All of these functions are handled automatically; in other words, a substantially constant temperature is maintained in the occupied space over any reasonable range of outside weather conditions.

While reverse cycle units have been tailor-made for specific jobs in a considerable number of cases, this is an example of the manufacture of a fixed design unit in quantity on a production line. In order to make such a unit practical over the considerable range of temperatures expected, a number of advanced and ingenious features have been incorporated.

Conditioned air is drawn to the unit through a typical return air system, Fig. 17, is filtered, and is taken through coils which may be either

Table 6 — Capacity Data for the Four Airtopia Units

Model	Compressor HP	Blower HP	Air Volumes, CFM	
			Conditioned Air	Outside Air
300	3	2 — ¾	1800	2400
500	5	2 — 1	3000	4000
750	7½	2 — 1½	4500	6000
1000	10	2 — 2	6000	8000

cold or hot, depending on the cycle called for by the thermostat. The blower above the coils distributes air to the various spaces through a duct system. At the same time, outside air is drawn in and over the outside air coils, which may be either hot or cold (always the opposite of the conditioned air coils). The outside air blower discharges into a duct leading outdoors at some point away from the intake.

Whether or not the compressor and outside air blower are operating, a definite quantity of fresh air is always drawn through the fresh air filters and mixed with the conditioned air return in the plenum chamber under the conditioned air coils. By dampering off a portion of the fresh air filters, the fresh air intake may be set at any point between 10 and 25%. (For reasons outlined later, it is undesirable to set fresh air below 10%.) Since fresh air is continuously being drawn in, the conditioned space operates at a slight positive pressure and unless wind pressure is excessive, the extra air will leak out through window cracks, chimney, bathroom and kitchen vents, etc. It will also be seen that no need exists for opening windows, since both circulation and ventilation are taken care of, and windows may be permanently sealed, thus reducing heat leakage.

In an air-to-air machine, there are several feasible methods of arranging the heat transmission coils:

a—Two coils only may be employed, coil *A* being either an evaporator or a condenser, and coil *B* being either a condenser or an evaporator. By means of either manual or automatic valves, liquid is switched to one coil and compressor discharge switched to the other. But since a well-designed evaporator may not necessarily work well as a condenser, it is likely that a design like this will not have high heat transmission factors. In a small room cooler, where extra surface can be added to the coils at low cost, this is not particularly important, and refrigerant switching has been successfully employed.

b—We again may employ two coils only, but coil *A* will always be an evaporator and coil *B* will always be a condenser. The conditioned air stream will be drawn over the evaporator during the cooling cycle and over the condenser during the heating cycle; the outside air stream will always be drawn through the *opposite* coil. If this is to be done automatically, a minimum of four motor driven dampers is required, and the resulting duct system is likely to be bulky and expensive. Also, since the outside air stream is generally the larger, both coils must be designed for the larger stream and a good balance is hard to obtain.

A variation of this plan is the "semi-regenerative" system, in which outside air and conditioned air return are mixed in equal quantities and the resulting mixture drawn through an evaporator and condenser placed side by side. By mixing vanes placed behind the coils, half the air (either cooled or heated) is sent to the conditioned air ducts and the remainder (either heated or cooled) is discharged outside. The high percentage of fresh air makes this plan applicable only in mild climates and more desirable in installations where heavy occupancy load, extreme smoking and the like, show the need for large quantities of makeup air.

c—To overcome the disadvantages of both systems described above, we may use two evaporators and two condensers, one of each in each air stream. Switching problems are now simplified; one evaporator and one condenser are chosen by one of several means, and the air stream and coil sizes may be balanced as required. However, we have raised the cost, since we have four coils instead of two, and each air stream must go through two coils so we have increased the power required to move the air.

d—Still a fourth plan, used in Airtopia, is to build an evaporator and a condenser in one assembly, on one set of fins. Two assemblies are required, one for the conditioned air stream and one for the outside air. Such a scheme is smaller and less expensive than four separate coils, the resistance to air flow is not as great, and considerable extra heat transmission is obtained by the pipes and extra fin surface of the inactive coil, which is of course attached to the active coil.

Condenser design has been carefully planned for low pressure drop and high mass velocity, in order that the highest possible heat transmission may result. When on the cooling cycle in desert weather, a well-designed air cooled condenser which will operate at a temperature difference of five degrees less than some other design means a reduction of condensing pressure of 10 lb, which means in turn a material reduction of power input. In the Airtopia unit, a multi-pass condenser is used, and uncondensed vapor is recirculated by means of an ingenious injector. In order to utilize finned surface to the greatest advantage, compressed vapor is put through a desuperheater coil before going to the main condenser.

The evaporator is also of multi-pass design, using a standard thermostatic expansion valve and a Drayer-Hanson

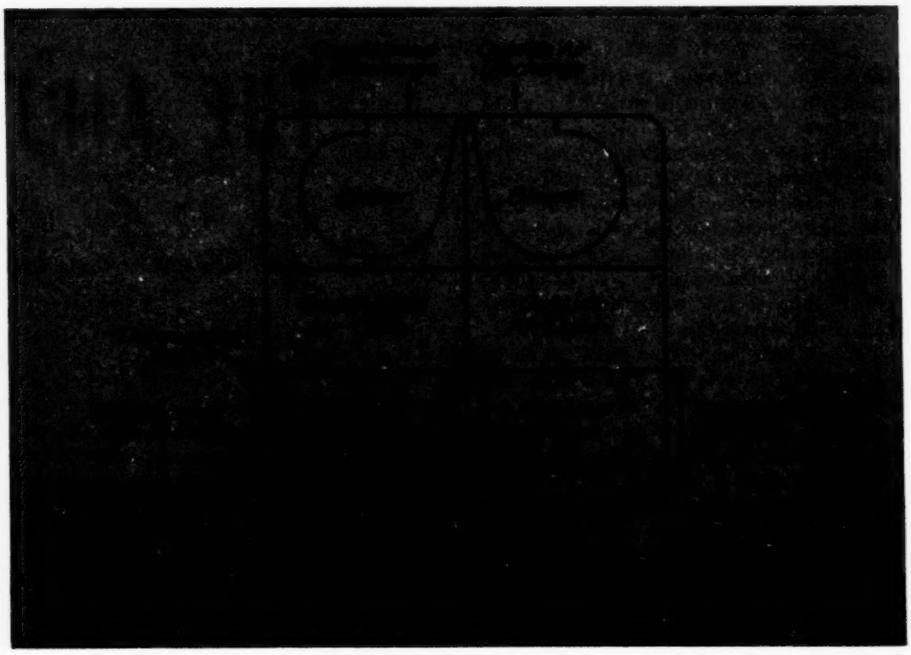


Fig. 17. Schematic arrangement of the component parts of the Airtopia unit showing the air flow.

liquid distributor which accurately meters refrigerant to the various air units.

The choice of cooling or heating cycle, determined by the thermostat, is effected by: 1. Solenoid valves in the liquid lines, the opening of one of which activates that evaporator; 2. By a switching valve of special design which, controlled by pilot solenoid valves, directs compressed vapor to the correct condenser.

The compressor unit is of conventional type, V-belt driven, using Freon-12.

Airtopia units are now being manufactured in four sizes. Capacity data for them appear in Table 6.

Model 300 uses a single speed compressor motor; other models use a 4/8 pole, 1740/870 rpm two-speed motor. All motors are 3-phase. In the larger models, the control circuit is such that the motor always starts on half speed, and is automatically shifted to full speed when the position of the thermostat modulating motor shows that the cooling or heating load cannot be carried at half speed.

The thermostat bulb is located in the conditioned air plenum immediately behind the conditioned air filters, and since air circulation is continuous, it reacts to the temperature of a mixture of air from all the conditioned spaces.

The temperature at the thermostat bulb produces a pressure in a bellows connected to the core of a solenoid. A change of flux in the solenoid coil actuates a small modulating motor, similar to a damper motor, so that cam switches close and open the circuits necessary to start and stop the motors. On the two-speed models, a typical range of temperatures corresponding to various cycles is given in Table 7.

Table 7 — Temperature Range

70F—full speed heating
71F—half speed heating
72F—stop heating
74F—stop cooling
75F—half speed cooling
78F—full speed cooling

On the single speed Model 300, the first and last positions do not exist, and heating is established at 71F and cooling at 75F. The range is adjustable by the user.

A time switch is provided so the unit can be turned off when the building is not occupied, and a "hand-off-automatic" switch can be operated by the owner to supersede the time clock if desired. Such a switch might be used in an office installation if overtime work were in progress.

In order to make an air-to-air unit practical at outside air temperature down to 10F, a fresh air preheater has been located as shown in Fig. 17, immediately behind the fresh air filters.

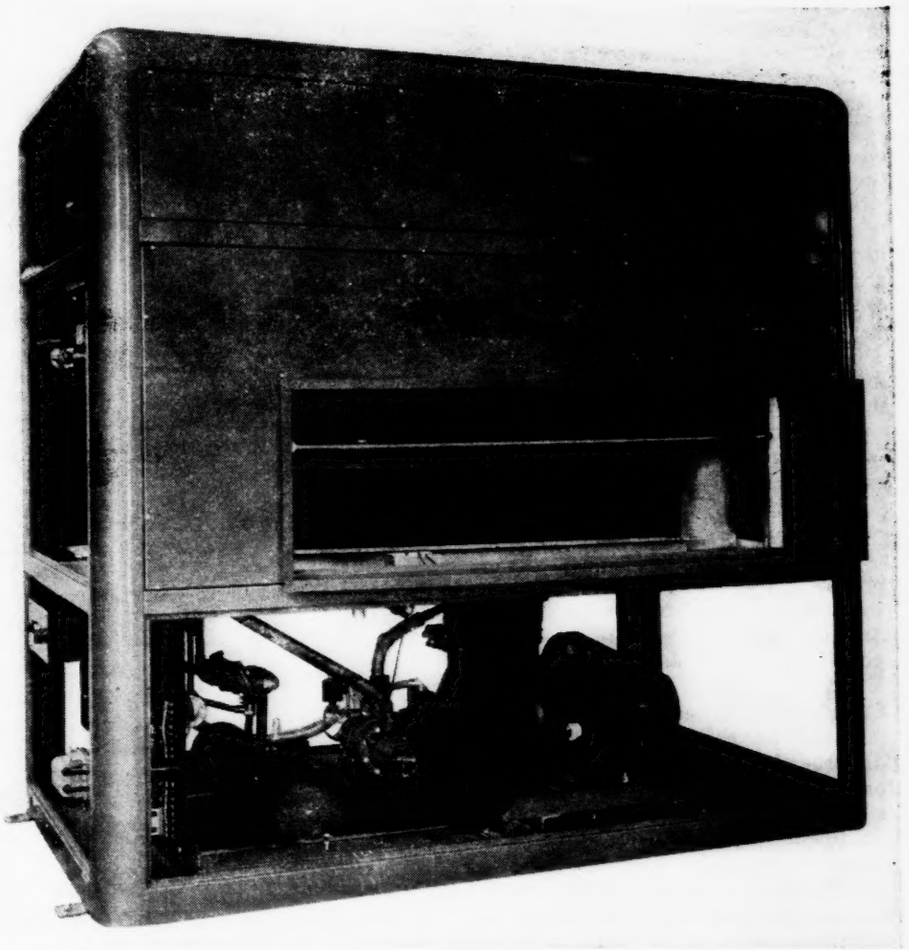


Fig. 18. Airtopia unit with the lower and front panels removed. A definite amount of air always passes through the fresh air filters.

This is a fin coil through which is circulated warm liquid from the receiver. In a typical example, the liquid temperature is reduced 43F while the fresh air temperature is increased 11F. The reduction of liquid temperature increases the refrigerating effect of the Freon-12 about 19% per lb, while at the same time, if a 20% fresh air addition is being used, there is an increase of temperature of the mixture of fresh air and return air under the conditioned air coil of 2.2F. Both these factors improve the capacity and performance of the unit on the heating cycle.

Use of this heat-exchanger is the principal reason why fresh air should not be reduced below 10%. Since the fresh air being brought in during the cooling cycle is likely to be about the same temperature as the warm liquid refrigerant from the receiver, this coil is not used in the cooling cycle.

Dehumidification on the cooling cycle is accomplished in the conventional way: By the impingement of the mixture of conditioned air return and fresh air against cold coils. The balance between air mass velocity and evaporator area is such that at high speed the typical relative humidity in the conditioned space is about 50%; slightly higher at half speed. However, in most installations no outside drain is connected; instead, the condensed moisture is conducted by a horizontal pipe to a point immediately under the outside air, or condenser, coils, and the moisture atomized by being picked up by a spinner disc running at a motor speed.

When the small particles of water strike the hot condenser surface, they are evaporated and carried away by the updraft of outside air. At the same time, the evaporation tends to cool the coils and reduce condenser temperature and power input to the compressor. No line will form, since the water being evaporated is actually distilled. In high humidity areas, a supplementary drain may be connected to handle water in excess of the amount which the air stream will absorb.

In one test installation in the southeast, as much as 25 lb (3 gal) of water per hour have been condensed, and a supplementary drain was connected.

On the heating cycle, moisture condensed on the outside air (evaporator) coils is conducted to a point under the conditioned air coils and an identical spinner breaks it up so it can be evaporated into the conditioned air stream, thus raising the relative humidity in the conditioned space, without need of an auxiliary water supply.

This is of course not "humidity control" in the usual sense; no humidistat is employed and the standard Airtopia

is not designed for process conditioning. Where exact relative humidity is required, supplementary equipment can be added for such applications.

The condensation of moisture on the outside air (evaporator) coils during the heating cycle will occur down to outside air temperatures of about 40F; below this temperature frost may form. If the machine is cycling, it is probable that frost will melt between cycles. If the load is such that the machine runs continuously, the frost may build up over a period of hours and reduce the fin spacing. Should this occur, an automatic switch will turn off the outside air fan and reverse the operation of the unit so that the frost will be melted by condenser heat. Normal operation is restored automatically as soon as the frost has melted.

Typical cooling and heating capacities for the standard model 500 are given in Table 8. Capacities for other models are in proportion to the horsepower.

Due to the automatic nature of the unit and its quick response to calls for either heating or cooling, the problem of estimating cost of operation is somewhat complex. The problem has been approached from these angles:

a—In weather requiring nothing but heating, the cost will vary with the difference between 65F and the actual outside temperature. This agrees with the usual degree-day calculations, and assumes that the night temperatures in the conditioned space will be somewhat lower than those held in the daytime.

b—In weather requiring nothing but cooling, the cost of operation is substantially that of any other air conditioning system, if proper weight is given to the absence of any cost for condenser water, and the somewhat lower coefficient of performance of an air-cooled machine.

c—In mild weather, where heating is needed in the morning and cooling in the afternoon, only experience in a given area will tell what the cost is likely to be, except that we know it will be lower than in the summer or winter.

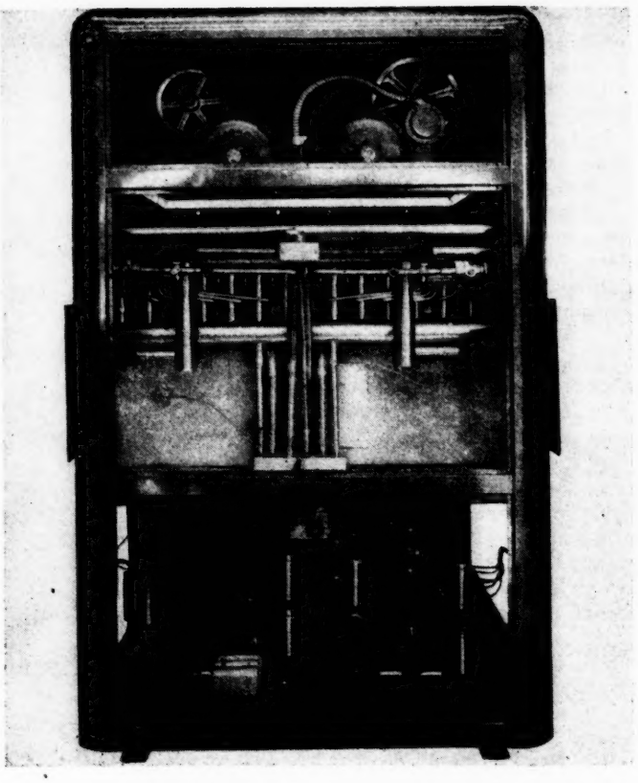


Fig. 19. Front view of Airtopia showing blower motors, coil assemblies, and electrical panel.

Cost of operation tests are now in progress, and some data may be expected by the end of another year. In the meantime, estimates in the Los Angeles area are about 2000 kwh per year per hp; in localities like Bakersfield, Calif., or Phoenix, Ariz., where both summers and winters are more severe, about 3,500 kwh per year per horsepower may be expected. These figures include substantial allowances for both conditioned air blower and outside air blower, and are based on the assumption that the unit is properly sized to the load.

A number of installations have been made in homes, doctors' offices, restaurants, malt shops, drug stores, small office buildings, and the like. While adjustments to fit individual conditions have been necessary on some of these jobs, the customers are markedly enthusiastic about the new freedom from weather worries that a year-around air conditioning system has brought them.

A number of interesting experiences have accompanied the first installations. One request for service came from a small commercial establishment with a newly installed Airtopia unit, when it was found that the return air filters were covered with dirt to a thickness of ¼ in. Upon inquiry, it was found that the unreasonable accumulation of dirt on the filters was due to the janitor sweeping all floor dirt into the return air duct. Someone had told him that the return air system acted just like a vacuum cleaner.

One doctor was particularly pleased because the adjustable thermostat enabled him to raise the temperature in his operating room during minor surgery, and to restore normal temperature just as easily afterward. We who are so familiar with thermostats sometimes fail to realize how important they can be to customers.

Distribution of Airtopia was originally limited to California and Arizona, but the flood of inquiries from other areas dictated an expansion of sales plans to include the entire south, and outlets are being negotiated as quickly as personnel permits. The market will become even wider when production of a line of water-to-air units is undertaken; such a unit will be feasible in climates where the winter water temperature does not fall below 40 to 50F, and where city water is low in cost, or where well water is easily available. Samples have already been built, and final design details are well along.

Table 8 — Heating and Cooling Capacities for Model 500

Cooling Capacity		Heating Capacity	
Outside Air F.	Btu per hr	Outside Air F.	Btu per hr
70	60,600	10	46,200
80	59,100	20	50,700
90	57,600	30	55,700
100	56,100	40	61,200
110	52,500*	50	67,300

*Special adjustment of compressor and outside air speed recommended for this temperature.

CANADIAN DEGREE DAYS (65F BASE) FOR RECENT HEATING SEASONS

The accompanying data for 25 important Canadian localities for the years 1940 to 1946 are presented through the cooperation of the Meterological Division, Air Services Branch, Department of Transport, Canada.

Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
CALGARY, ALTA.											
1940-41	279	263	1374	1209	1463	1221	1147	624	496	186	8262
1941-42	558	688	933	1370	1150	1406	1060	732	558	336	8791
1942-43	387	657	1320	1525	1885	1092	1438	624	583	396	9907
1943-44	342	620	864	1063	1240	1334	1308	648	391	294	8104
1944-45	252	477	1098	1302	1426	1305	1057	1029	567	364	8877
1945-46	492	670	1464	1566	1333	1201	949	612	536	326	9149
CHARLOTTETOWN, P. E. I.											
1940-41	174	626	828	1308	1485	1224	1265	819	546	252	8527
1941-42	312	589	822	1141	1463	1305	1042	774	360	117	7925
1942-43	132	453	858	1414	1634	1182	1197	870	533	294	8567
1943-44	144	428	804	1401	1364	1386	1283	840	381	174	8205
1944-45	152	558	750	1265	1355	1246	1091	762	583	271	8033
1945-46	189	552	825	1271	1445	1327	1020	849	468	183	8129
CRESCENT VALLEY, B. C.											
1940-41	105	570	1038	1100	1091	924	725	438	332	120	6443
1941-42	420	642	852	1097	1308	991	868	513	418	249	7358
1942-43	270	623	1026	1135	1553	938	1073	546	471	264	7899
1943-44	264	620	891	1194	1234	1038	989	537	341	126	7234
1944-45	267	508	888	1215	1116	997	933	717	322	216	7179
1945-46	412	636	960	1159	1200	966	831	582	288	214	7248
EDMONTON, ALTA.											
1940-41	288	682	1530	1575	1863	1490	1265	564	459	147	9863
1941-42	594	725	1041	1649	1352	1417	1073	780	468	264	9363
1942-43	426	676	1200	1882	2080	1282	1541	645	515	306	10553
1943-44	366	657	924	1249	1466	1502	1364	555	360	234	8677
1944-45	357	586	1251	1516	1686	1436	1147	987	477	240	9683
1945-46	510	747	1626	1792	1327	1579	1076	624	471	282	10034
FORT WILLIAM, ONT.											
1940-41	318	701	1272	1569	1767	1548	1420	762	440	168	9965
1941-42	330	685	1092	1404	1631	1484	1141	696	552	240	9255
1942-43	432	725	1176	1798	2012	1532	1612	903	564	234	10988
1943-44	456	694	1212	1649	1414	1630	1460	906	465	231	10117
1944-45	325	663	951	1600	1934	1512	1079	936	682	294	9976
1945-46	417	794	1164	1730	1761	1683	1079	837	589	343	10397
GRAND PRAIRIE, ALTA.											
1940-41	No data available										
1941-42	627	713	1086	1637	1197	1310	1153	840	415	255	9233
1942-43	390	688	1560	2015	1990	1266	1519	660	577	306	10971
1943-44	423	713	1020	1221	1507	1534	1327	594	397	222	8958
1944-45	375	611	1320	1565	1717	1481	1227	1008	465	267	10036
1945-46	477	772	1767	1851	1671	1714	1228	780	400	282	10942
HALIFAX, N. S.											
1940-41	174	570	735	1079	1333	1053	1104	720	561	264	7593
1941-42	246	502	738	1042	1240	1137	905	714	440	192	7156
1942-43	72	375	756	1246	1401	1044	1051	813	496	408	7662
1943-44	143	391	678	1187	1175	1201	1113	762	378	222	7250
1944-45	130	474	681	1070	1178	1095	927	642	572	285	7054
1945-46	163	504	758	1162	1262	1173	861	786	470	238	7377
LONDON, ONT.											
1940-41	180	555	855	1150	1345	1221	1209	468	223	0	7206
1941-42	66	434	774	1048	1339	1310	1014	480	254	0	6719
1942-43	180	490	846	1321	1466	1165	1116	825	356	57	7822
1943-44	192	564	882	1252	1190	1218	1150	747	149	62	7406
1944-45	172	519	798	1376	1624	1190	701	564	515	165	7624
1945-46	181	577	798	1380	1296	1215	732	654	353	133	7319
MEDICINE HAT, ALTA.											
1940-41	87	459	1428	1228	1507	1232	1088	540	291	60	7920
1941-42	438	589	885	1290	1271	1506	998	594	412	228	8211
1942-43	318	617	1260	1513	2071	1249	1485	468	440	222	9643
1943-44	264	499	915	1184	1336	1322	1364	504	220	153	7761
1944-45	238	428	1263	1466	1550	1394	1011	846	453	205	8854
1945-46	398	577	1320	1655	1442	1285	775	468	434	152	8506
MONCTON, N. B.											
1940-41	270	725	930	1392	1593	1215	1234	798	508	216	8881
1941-42	302	620	882	1265	1531	1324	1029	750	316	90	8109
1942-43	174	701	966	1507	1668	1221	1181	900	493	282	9093
1943-44	234	496	891	1553	1420	1433	1296	876	341	192	8732
1944-45	200	620	822	1392	1494	1310	1097	684	570	265	8454
1945-46	258	651	903	1370	1513	1383	998	909	462	194	8641
MONTREAL, P. Q.											
1940-41	147	608	948	1364	1637	1294	1259	462	223	0	7942
1941-42	153	564	867	1293	1547	1341	973	576	155	0	7469
1942-43	144	471	900	1534	1770	1288	1290	888	321	53	8659
1943-44	257	546	918	1500	1730	1404	1252	768	108	73	8556
1944-45	198	558	879	1590	1841	1305	874	522	403	112	8282
1945-46	187	580	1032	1479	1574	1542	848	687	362	114	8405
NORTH BAY, ONT.											
1940-41	330	763	1170	1500	1817	1498	1488	666	387	54	9673
1941-42	279	707	1038	1476	1742	1532	1184	672	403	105	9138
1942-43	336	629	1116	1699	1903	1478	1544	1065	440	90	10300
1943-44	378	672	1098	1631	1457	1543	1485	966	279	99	9608
1944-45	300	676	1005	1587	1987	1389	967	735	570	222	9438
1945-46	368	753	1044	1665	1705	1708	967	816	546	223	9795

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CANADIAN DEGREE DAYS (65F BASE) FOR RECENT HEATING SEASONS

Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
OTTAWA, ONT.											
1940-41	177	660	999	1476	1748	1425	1392	534	267	0	8678
1941-42	162	589	918	1401	1631	1422	1029	552	229	0	7933
1942-43	183	512	960	1600	1854	1394	1352	930	332	57	9174
1943-44	302	558	945	1562	1466	1493	1339	804	121	0	8590
1944-45	215	595	900	1612	1894	1338	877	546	471	120	8568
1945-46	244	598	957	1538	1634	1568	834	696	384	146	8599
PENTICTON, B. C.											
1940-41	———		No data available			———		411	285	60	756
1941-42	312	515	741	936	1190	851	784	480	341	135	6285
1942-43	174	477	846	1035	1457	902	911	456	384	156	6798
1943-44	195	508	774	1066	1125	977	868	492	285	24	6314
1944-45	168	441	792	1122	1023	865	790	588	260	120	9169
1945-46	281	539	858	1017	1035	924	753	522	198	163	6290
PORQUIS JUNCTION, ONT.											
1940-41	375	849	1362	1717	2049	1641	1621	870	434	126	11044
1941-42	348	812	1242	1693	2003	1666	1324	795	493	99	10475
1942-43	456	747	1242	1996	2080	1646	1708	1134	496	261	11766
1943-44	468	769	1224	1891	1590	1757	1649	1104	403	198	11053
1944-45	371	806	1146	1730	2251	1537	1091	894	698	260	10804
1945-46	445	871	1206	1857	1910	1938	1091	987	691	258	11254
PRINCE GEORGE, B. C.											
1940-41	297	685	1335	1246	1324	1176	862	528	434	192	8079
1941-42	498	657	954	1386	1290	1142	1042	681	388	222	8260
1942-43	330	682	1272	1423	1804	1042	1224	654	589	273	9293
1943-44	420	713	951	1215	1265	1201	1166	636	459	210	8236
1944-45	405	660	1011	1590	1414	1193	1042	894	440	303	8952
1945-46	504	769	1383	1352	1329	1204	992	783	397	276	8989
QUEBEC CITY, P. Q.											
1940-41	234	676	1014	1485	1686	1316	1259	627	329	51	8677
1941-42	258	648	990	1361	1680	1358	1073	729	248	42	8387
1942-43	186	533	984	1637	1779	1347	1364	978	431	128	9358
1943-44	270	586	966	1624	1494	1517	1333	870	260	127	9047
1944-45	186	629	870	1550	1755	1302	1011	612	465	156	8536
1945-46	230	663	987	1510	1637	1523	980	822	409	154	8915
REGINA, SASK.											
1940-41	138	536	1485	1686	1903	1711	1426	732	322	66	10005
1941-42	426	744	1164	1600	1612	1557	1252	723	459	249	9786
1942-43	426	698	1329	1841	2251	1557	1733	666	561	252	11314
1943-44	372	608	1092	1432	1538	1682	1655	675	276	198	9528
1944-45	315	583	1278	1680	1820	1641	1147	945	604	290	10303
1945-46	489	769	1533	1897	1857	1694	1194	600	527	202	10762
ST. JOHN, N. B.											
1940-41	258	660	852	1246	1472	1120	1156	732	484	264	8244
1941-42	279	577	810	1166	1435	1260	942	726	434	228	7857
1942-43	216	484	876	1401	1544	1159	1138	864	512	258	8452
1943-44	219	484	822	1401	1311	1317	1172	804	417	297	8244
1944-45	191	555	792	1276	1432	1190	958	678	546	282	7900
1945-46	244	570	840	1293	1401	1291	899	834	496	261	8129
SASKATOON, SASK.											
1940-41	177	589	1578	1693	1947	1674	1398	693	391	54	10194
1941-42	471	725	1152	1643	1590	1544	1128	756	465	198	9672
1942-43	423	676	1512	1996	2275	1518	1742	654	570	258	1624
1943-44	372	608	1068	1519	1668	1737	1711	645	291	180	9799
1944-45	292	574	1320	1668	1879	1641	1135	1038	598	283	10428
1945-46	507	756	1572	1906	1887	1764	1138	600	487	215	10832
TORONTO, ONT.											
1940-41	154	538	814	1075	1305	1136	1159	442	233	40	6896
1941-42	118	426	692	1020	1248	1233	906	470	289	56	6458
1942-43	179	431	766	1240	1403	1105	1098	795	329	50	7396
1943-44	231	531	818	1192	1093	1142	1135	739	205	70	7156
1944-45	142	483	756	1210	1549	1093	691	545	468	140	7077
1945-46	164	524	752	1252	1208	1189	698	621	346	118	6872
VANCOUVER, B. C.											
1940-41	135	363	744	738	725	636	552	420	335	156	4804
1941-42	264	434	603	784	868	694	688	456	322	189	5302
1942-43	239	397	702	756	1057	694	735	462	378	347	5767
1943-44	198	428	612	856	797	731	701	456	347	159	5285
1944-45	272	403	588	865	760	669	701	564	279	186	5287
1945-46	288	459	714	825	781	678	673	525	267	192	5402
VICTORIA, B. C.											
1940-41	186	363	660	666	657	538	496	390	304	210	4470
1941-42	267	403	534	707	738	644	620	432	316	210	4871
1942-43	192	356	630	685	936	596	657	420	409	252	5133
1943-44	126	369	540	704	707	644	642	456	326	234	4748
1944-45	190	301	549	735	685	605	648	546	291	228	4778
1945-46	270	388	660	735	716	627	636	504	285	243	5064
WINDSOR, ONT.											
1940-41	———	No data available			———	1209	1114	1079	390	105	0	3897
1941-42	0	300	711	952	1234	1193	856	414	186	0	5846
1942-43	96	397	762	1262	1370	1056	992	702	298	0	6935
1943-44	147	462	852	1184	1107	1102	1066	687	87	0	6694
1944-45	106	312	705	1277	1507	1081	601	468	440	134	6631
1945-46	117	477	735	1271	1184	1078	617	528	282	98	6387
WINNIPEG, MAN.											
1940-41	93	502	1338	1618	1879	1702	1476	687	260	18	9573
1941-42	342	713	1242	1606	1658	1669	1159	720	496	111	9716
1942-43	378	601	1320	1972	2257	1663	1683	756	510	202	11342
1943-44	371	577	1170	1600	1612	1815	1612	726	279	144	9906
1944-45	294	660	1116	1705	1894	1610	1104	978	626	185	10172
1945-46	413	747	1356	1931	1975	1800	1091	627	453	88	10481

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Bus Heating

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The problem of bus passenger comfort and how it has been approached by General Motors is discussed in the accompanying article which is an abstract of a much lengthier paper entitled "Automotive Space Conditioning" presented by the authors before the National Transportation and Maintenance meeting of the Society of Automotive Engineers, held in Chicago, October 16-17.

PASSENGER comfort is determined, mainly, by three conditions: a feeling of pleasure upon entering the coach followed shortly by an unawareness of the temperature, complete absence of odors, and good visibility even in the coldest winter weather. Although these conditions may seem to be unrelated, their solution can be achieved by processing an adequate supply of fresh air and distributing it in a manner to be described later. The cases of unpleasant temperatures, drafts, odors, and fogged or iced windows are known and their elimination is not too difficult or costly.

An understanding of the design conditions is a prerequisite to the solution of the problem of producing correct temperature, freedom from odors, and clear glass in automotive vehicles. We say "problem" because a single solution can be made to meet all these difficulties. It is the purpose of this paper to present the above-mentioned design conditions and to show how a single treatment effectively eliminates the three undesirables—extreme or varying temperatures, obnoxious odors, and a hazardous and irritating lack of visibility due to fogged or iced windows.

It will be shown that auxiliary heat is the answer to the problem. Experience in the design of automotive heating systems shows that the inadequacy of rejected heat from the engine is the fundamental cause of temperature, odor, and visibility complaints. Conversely, it will be seen that an adequate heat supply makes possible abundant fresh air properly tempered, odor-free, and suitable as an agent to keep all glass clear at all times. The hazard of driving with fogged glass has been experienced by every winter driver. Its elimination, obviously, is of paramount importance. The facts underlying inadequate heat from the engine and the results of this inadequacy are presented as follows:

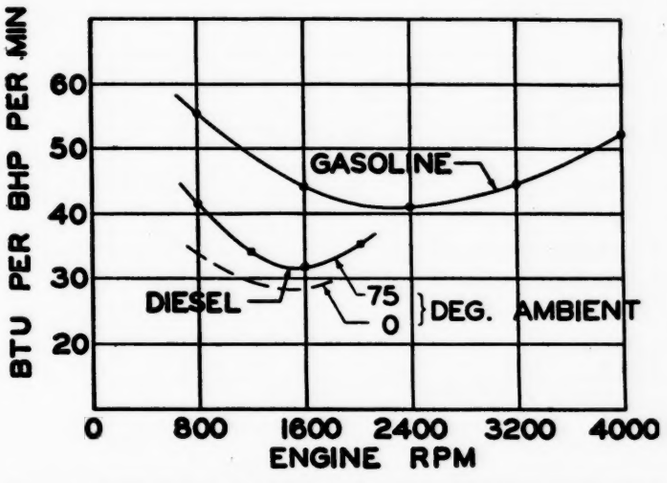


Fig. 1. Heat rejected to cooling water, gasoline and diesel engines, full throttle.

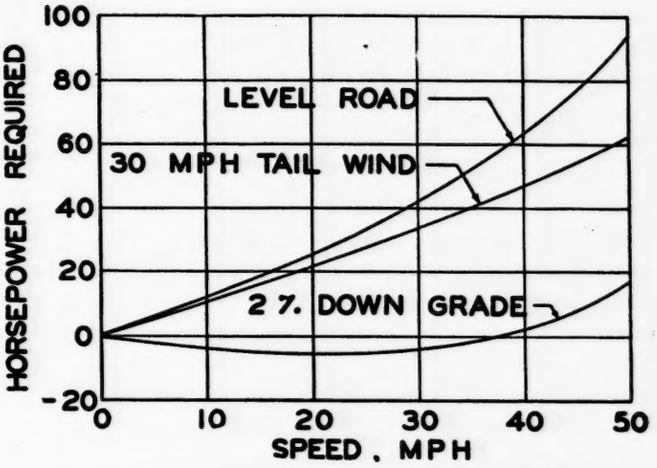


Fig. 2. Power-speed curve, coach at 29,000 lb gross.

Space heating designers are agreed that —30F is a design condition for an important part of the northern United States and southern Canada. Automotive designers have not accepted this as the design temperature for car heaters because of the following apparent reasons: They assume that riders will be clothed for outdoor activity; they know that heat output from the automobile or coach engine is entirely insufficient under some operating conditions; they know that under favorable operating conditions engine heat is adequate if no fresh air is provided and if the vehicle body is substantially air-tight. That engine heat is far from adequate under normal operating conditions is shown by Fig. 1, 2, and 3.

Fig. 1 shows approximately the heat rejected to the cooling system per minute per brake-horsepower, some part of which is available to the car heater. Coach heating installations in the past have utilized available heat from the engine in various ways, but the results have not been satisfactory because the engine does not reject enough heat under some design operating conditions.

The two curves shown on Fig. 1 represent data

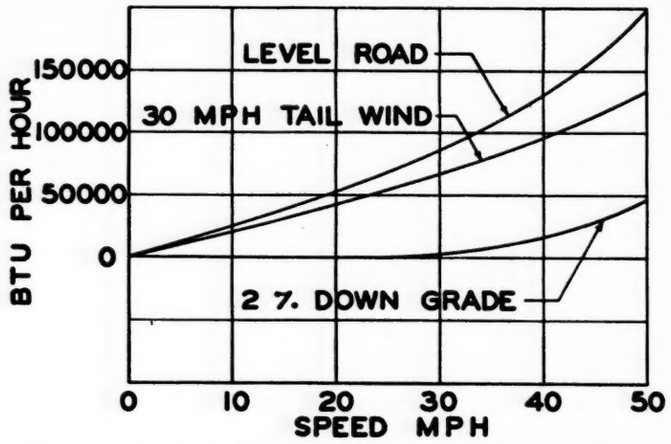


Fig. 3. Heat available for comfort heating from GM 6-71 diesel engine.

obtained on heat rejection to the cooling water for an average of ten different General Motors gasoline engines and for a GM 6-71 diesel engine. Heat rejection in Btu per minute per brake-horsepower is plotted against engine rpm (full throttle) and shows rejection to be lowest near normal operating speeds for both engines. For our calculations we have used a diesel engine as the typical coach engine and considered a heat rejection of 35 Btu per minute per brake-horsepower to be an average value for the range of speeds most likely in normal operation. As a matter of interest it is noted that the heat loss to the cooling system is only slightly less than the equivalent useful output of the engine. Data for the gasoline curve were taken from an average of ten General Motors Coach engines with displacement varying between 216 and 431 cu in.

Fig. 2 shows the brake-horsepower required to operate a 45-passenger coach at varying road speeds.

The curve of engine output is obtained from use of the formula $R_w = V^2/10$ for the force necessary to overcome wind friction, and the formula $R_r = .012\ W$ for the force required in overcoming road friction. The upper curve shows the horsepower output of a fully loaded parlor type coach moving along a level road through still air at various speeds, while the curve immediately below shows the effect of a 30 mph tail wind upon the same coach. The lower curve is for the same coach moving down a 2% slope through still air at varying speeds. It is assumed that the overall transmission efficiency is 85%.

Formulas used in plotting these curves were: Rolling resistance, $R_r = 0.012\ W$, and wind resistance, $R_w = 0.1V^2$, where W = weight of coach in pounds, and V = coach velocity in mph.

For example, assume a fully-loaded coach weighing 29,245 lb, still air, and level road, moving at 10 mph. Then $R_r = .012 \times 29{,}245 = 351$ lb; and $W_r = .1 \times (10)^2 = 10$ lb; Horsepower equals

$$\frac{361 \times 14.6}{550 \times .85} = 11.25.$$

For 30 mph tail wind, level road, and 10 mph, $R_r = 351$ lb, $R_w = -40$ lb and Horsepower = 9.71. For still air, 2% down grade, and 10 mph, Horsepower = —4.25.

Fig. 3, derived from Fig. 1 and Fig. 2, shows that heat available from the coach engine under some operating conditions may approach zero. Fig. 3 shows the heat rejection to the cooling system under the conditions outlined on Fig. 2. The output represented by the lower curve is due solely to the friction horsepower of the engine turning over at a speed corresponding to the coach speed but developing no brake-horsepower.

Heat is lost from a properly conditioned coach primarily by transfer through the walls and by the large volume of heated air which must be exhausted. The heat loss through the structure varies greatly with type of construction as well as with the weather. The development of the General Motors system has been conducted in a simulated coach and a pre-war 40-passenger transit coach at the GMC Truck and Coach plant in Pontiac, Mich. The results obtained in our coach mock-up and test vehicle are correct only for this vehicle. It is the problem of each engineer to determine the heat loss through the structure for his own design, but our experiments will prove of value in suggesting means of determining procedure.

Formulas used in and conditions of Fig. 3. Horsepower × (Btu per minute per brake-horsepower) × 60 = Heat available in Btu per hour. Then, for still air, level road, and at 10 mph, Heat available = 11.25 × 35 × 60 = 23,600 Btu per hr. For a 30 mph tail wind, level road and 10 mph, Heat available = 20,400 Btu per hr. In the case of still air, 2% down grade, and 10 mph, the clutch will be engaged and the engine will be turning over at a speed corresponding to that of the coach. Assume outside air at —30F, coach speed at 30 mph, exhaust temperature at 100F, blower displacement at 300 cfm to the engine air box. Then the enthalpy of air at —30F = —7.053 Btu per lb, and the enthalpy of air at 100F = 24,029 Btu per lb. Therefore, 31.082 Btu are added per pound of intake air, a volume of air at —30F of 10.822 cu ft per pound.

$$\frac{300}{10.822} = 27.7 \text{ pounds of air intake per min.}$$

27.7 × 31.082 = 862 Btu added per min. to heat the air.

Friction horsepower = 19.1.

19.1 × 42.41 = 810 Btu per min.

Since the engine is giving up about 860 Btu each

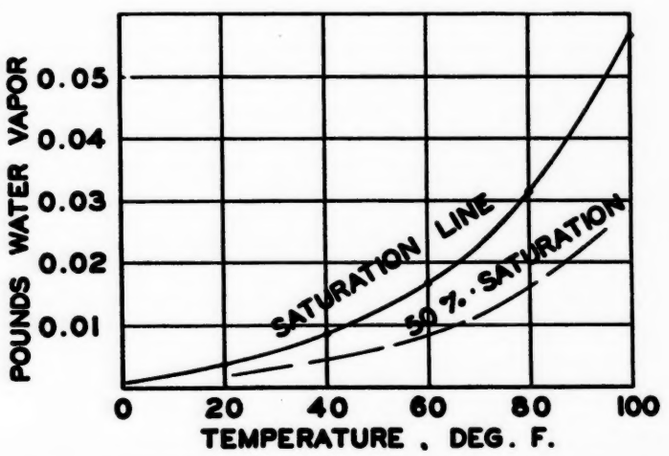


Fig. 4. Moisture holding capacity of 20 cu ft of air.

minute to heating air and the heat equivalent of the friction horsepower is about 810 Btu, no heat will be given up to the cooling water and the diesel engine is actually air cooled.

The 2% down grade and high tail winds occur with sufficient frequency and length of duration as to cause objectionable failure of the heating system and, inevitably, drivers will attempt to correct this condition. Such corrective measures usually include closing of fresh air inlets or stopping of fresh air blowers, or both, and once this has been done it usually remains that way.

These periods of low power requirements result in simultaneous periods of discomfort which, in turn, are the design conditions underlying the prevalent type of car heating and body seals. To alleviate this condition, designers have resorted to recirculating the enclosed air to conserve heat, and to sealing the bodies to prevent infiltration of cold air.

It should be noted that a pressurized interior would prevent infiltration of unheated air. Pressurization, however, requires an abundant supply of fresh air forced into the body and, therefore, an abundant supply of heat at the fresh air intake.

The effect of resorting to recirculated air is to increase the amount of moisture in passenger space to

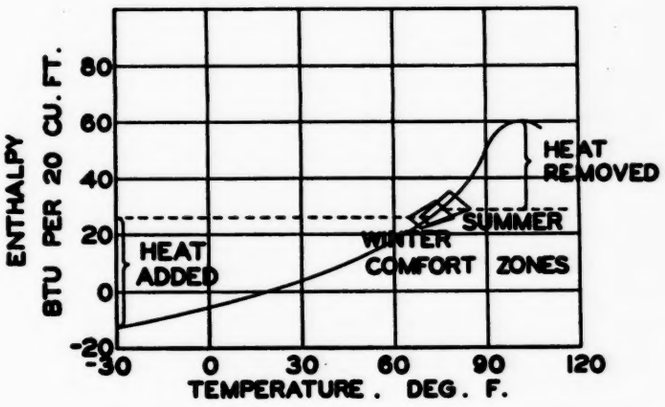


Fig. 5. Heat to be added or removed for comfort.

and beyond the saturation point because the occupants are giving off from one-quarter to one-third of a pound of moisture per person per hour. The amount of moisture that can be held as vapor by air is shown in Fig. 4. Airborne moisture in excess of this amount must condense as fog or dew. When warm air at a high degree of saturation comes into contact with cold window glass it must give up that quantity of moisture as fog or dew which it contains above the new saturation point for the temperature resulting from contact with the glass. The result is fogging and, in sub zero weather, instantaneous frosting.

The main curve is drawn to show the pounds of moisture per 20 cfm of saturated air at temperatures from 0F to 100F and standard atmospheric pressure. The dotted line under the main curve is drawn for 50% saturation. Data can be taken directly from tables and multiplied by the ratio of 20 cu ft to the volume of saturated air per pound dry air for the particular temperature under consideration to obtain the number

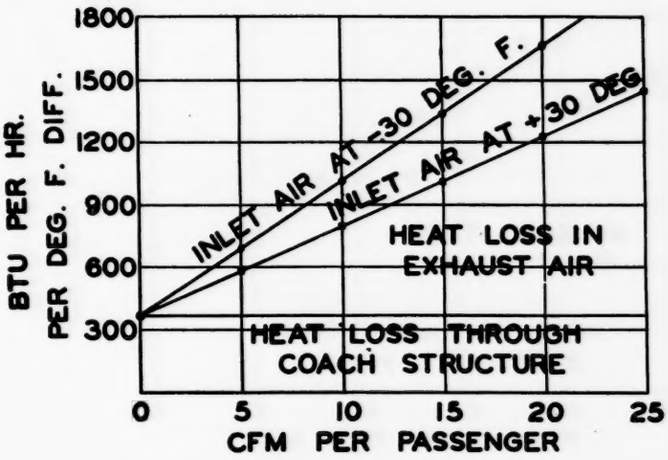


Fig. 6. Total heat required for a typical 45-passenger coach, empty.

of pounds of water vapor per 20 cubic feet of air.

Thus it is seen that occasional low power operation resulting in low heat supply necessitates either a continuous recirculation of passenger space air with its aforementioned objections or an auxiliary source of heat.

Another by-product of inadequate heat is obnoxious odor. As recirculated air accumulated moisture, so does it gather organic matter, perceptible as odors. Because recirculation of air fails to pressurize the passenger space, engine fumes are not kept out in cases where leakage exists between passenger and engine compartments. Engine fumes are one of the chief passenger complaints.

A third evil inherent in recirculation of heated air is unsatisfactory temperature control. Regulation and compensation for extreme temperature changes are necessary because the heating load will vary with suddent changes in passenger load, solar heat, and power output. Even with thermostatic control, temperature regulation is limited to heat loss through the body structure. The reader has possibly experienced the uncomfortable temperature changes resulting from the rapid filling of a theatre, banquet room or motor coach.

The question of what temperature is required to keep passengers happy is another problem requiring judgment on the part of the engineer. We have charts and data concerning the winter comfort zone (see Fig. 5) but should realize that warmly dressed passengers will be happier when the temperature is below that encompassed by the winter comfort zone. In the transit type bus the average passenger would prefer keeping his outer clothing on during the ride rather than removing it for only five or ten minutes. The problem now is to decide what temperature differential must be maintained between inside and outside air to keep both the passengers and coach operators satisfied. It is the opinion of the author that an inside temperature of about 45F would be comfortable in —30F weather, and an inside temperature of 55F in zero weather.

Now, let us consider the means of eliminating these undesirables. Temperature in the passenger space is stabilized, odors and cold drafts are eliminated, and

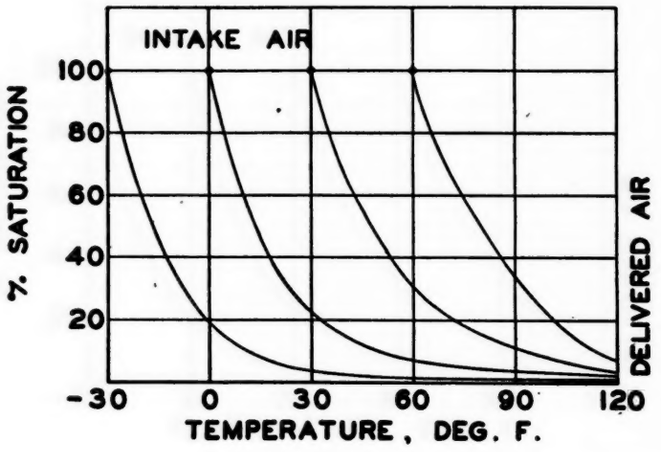


Fig. 7. Drying effect of heated air.

window glass is kept clear by one simple process, that of supplying a minimum of 20 cfm per passenger of fresh air to which the correct amount of heat has been added, prior to correct entry into the passenger space. Fig. 5 shows the amount of heat to be added to air in winter or subtracted from air in summer to obtain human comfort. The unit of air employed in this figure is 20 cu ft because this amount is accepted as the minimum needed to carry away interior odors and prevent infiltration of exterior odors. We have here calculated the enthalpy of air at —30F and 80% relative humidity and at 95F and 51% relative humidity. These conditions are from design wet-dry bulb data for the United States. We then plotted the summer and winter comfort zones on the graph and connected our known points with a smooth curve.

Fig. 6 shows the total heat required for a 45-passenger coach in Btu per hour per degree temperature difference inside to outside of coach for amounts of fresh air varying from 0 to 25 cfm per passenger. Fig. 6 is drawn for coach with no passengers; therefore, for each passenger we must subtract about 225 Btu per hour as the sensible heat he would give off in one hour. We have considered the heat given off per hour per person as equal to published figures for a person doing light work in a standing position since the average transit rider has been walking or expending energy in some manner prior to boarding the coach.

It must be noted, however, that air is being continuously exhausted from the coach at comfort temperature level and, therefore, carrying away heat. Heat is also being lost by conduction through the coach structure, as shown in Table 1, and must be replaced by an additional amount, also prior to entry into the coach space.

The heat requirement, as shown on Fig. 6, must be varied to compensate for change in passenger load, ambient temperature, and solar heat, the latter, of course, affecting the rate of heat loss through the coach structure. The means employed in the 1946 GM coaches to automatically compensate for changes in passenger load, ambient temperature, solar heat, engine temperature fluctuation, water flow, and other variants, is subject matter for a separate discussion. It must be noted here, however, that the continuous flow of 20 cfm per passenger of fresh outside air constitutes a conveyor by means of which a greater or lesser amount of heat can be quickly added or subtracted, thereby providing the essential means of controlling temperature within the passenger space. If the coach rapidly fills with passengers, then sun suddenly comes out from behind a heavy overcast, steep grades are encountered, or high speed occurs, the heat balance is violently disturbed and these changes must be compensated for instantly. A coach or an automobile may enter a long down grade, encounter strong tail winds, or be required to sustain low speed, as in heavy traffic. Compensation for the resulting cessation of heat supply cannot be accomplished by thermostatic control unless auxiliary heat is available to be quickly added to the rapidly moving air flow. Table 1 shows the procedure employed in determining heat loss through a typical coach structure.

The height of ordinate under the horizontal line in Fig. 6 was found from our experiments on a mock-up of a 20-passenger coach and represents the amount of heat transfer through a coach body for each degree of temperature difference between inside and outside conditions.

The slope of the lines to show heat loss in exhaust air was determined by finding the enthalpy change of a cubic foot of saturated air in being heated from

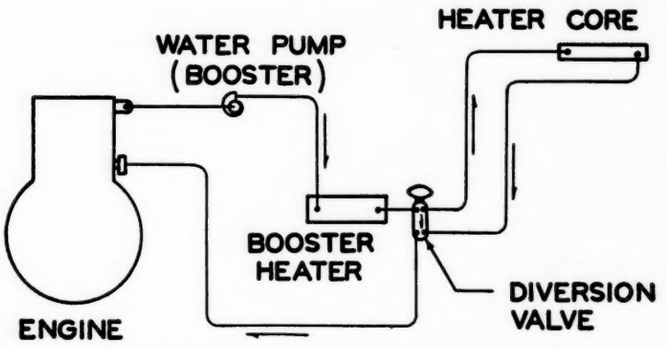


Fig. 8. Schematic diagram of heater installation.

—30 to —29F and comparing it with the change in enthalpy between + 30 and + 31F.

Fig. 7 shows the drying effect of raising the temperature of cold air. It is upon this principle that the GM system depends for clear windows. Four typical histories are outlined on Fig. 7, each showing the rapid decrease in per cent saturation as air is heated from —30, 0, + 30, and + 60F to higher temperatures. To calculate the per cent saturation at any new temperature we took the amount of moisture it actually contained (the amount held when saturated at the initial point) and divided it by the amount it could hold if saturated at the new temperature. By heating this cold air at the point of intake it enters the coach capable of absorbing any available moisture with which it comes in contact. If this air is directed over the window glass it will carry away any moisture clinging thereto or prevent moisture from condensing thereon. Thus, we interpose a dry blanket of air between the glass and the more humid interior air.

The increasing ability of air to hold moisture as its temperature increases is shown on Fig. 4.

In a coach or an automobile the engine supplies enough heat for proper ventilation only a part of the time. During the remaining time, when engine heat is inadequate, present style heating systems are limited to recirculation of vehicle air to conserve heat and foster the obsolete idea that heating and ventilating are separate arts.

It is to be concluded from the foregoing discussion that some source of auxiliary heat is required, which brings us to the subject of booster heaters, development of which during the past two years has been completed for GM coaches.

Prior to development of the fuel burning heater one of our diesel coaches was equipped with a water-jacketed exhaust manifold and a muffler containing finned tubing. The engine water extracted heat from the exhaust gases both in the exhaust manifold and in the muffler. Road trials of this coach in cold weather were rather disappointing in that only 7,500 Btu were recovered per hour with the engine operating near full load, while the exhaust gases actually extracted heat from the engine water during periods of low engine output. Our search, therefore, led to a self-contained, separately-fired booster heater. There were various makes of gasoline-fired air heaters in use, but to meet our requirements it was considered necessary to develop a heater which would burn either diesel fuel or gasoline.

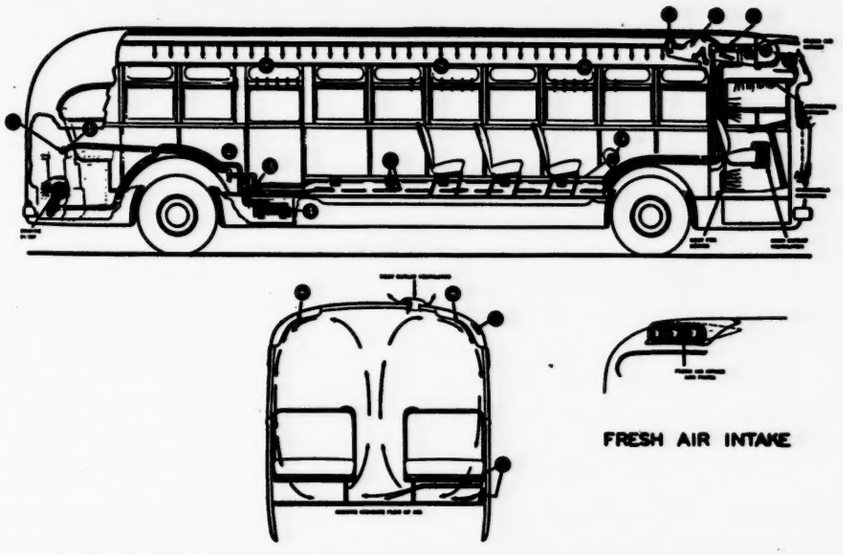


Fig. 9. Coach heating and ventilating, diagram (top), air flow diagram (bottom), and fresh air intake detail.

Table 1 — Experimental Data from Typical 20-Passenger Coach

1. Heat Supplied to Air.	
Water rate, gpm	15.25
Water rate, lb per hr.	2569
Water temperature entering, F	163.5
Water temperature leaving, F	154.8
Mean specific volume, cu ft per lb	0.0164
Heat from water, Btu per hr	22350
Electrical heat, Btu per hr	1975
Passenger heat, Btu per hr	0
Total heat input, Btu per hr	24325
2. Heat Loss in Exhaust Air.	
Outside dry bulb, F	58.5
Outside wet bulb, F	45.5
Barometric pressure, in. of Hg.	29.02
Outside air density, lb per cu ft	0.07411
Air flow into coach, cfm	435
Air flow into coach, lb per hr	1934
Average coach temp., F	97
Air temperature difference, F	38.5
Heat supplied per lb of air, Btu	9.32
Heat loss in exhaust air, Btu per hr	18020
3. Coefficient of Heat Transfer for Structure.	
Heat Loss Through Structure:	
Total heat supplied	24,325
Heat loss in exhaust air	18,020
Difference, Btu per hour	6,305
Coefficient of Heat Transfer for Body Structure:	
Heat loss through structure	6,305 Btu
Temperature difference inside to outside	38.5 F

$$\text{Coefficient} = \frac{6305}{38.5} = 164 \text{ Btu per hr per F}$$

In addition, a water-jacketed unit was not available, nor were controls available for the sequence and safeguards required in operation.

During the war, the Surface Combustion Corporation manufactured gasoline-burning aircraft heaters. Their Janitrol liquid fuel-burning heaters utilizing the whirling flame principle has proven outstandingly successful in many types of aircraft so that it was decided to convert their heater to a diesel fuel-burning unit with suitable controls for coach use. The basic principles of the aircraft heater were applied to the problem of a water-jacketed unit capable of burning either diesel fuel or gasoline. The automatic controls and safety devices employed in the aircraft heater were not suited to coach requirements. Surface Combustion and GMC engineers have collaborated in developing the jacketed type heater with controls and safety devices to meet the specialized requirements which obtain in coach operation.

One of the most important design factors of a heater for coach use is size. In our transit type coaches the heater is under the floor and the height of the heater had to be limited to about 1 ft to allow adequate road clearance. Location of body strength members under the floor restrict the other two dimensions to 1½ ft, and 2½ ft, respectively. In this restricted space we had to install a heater of adequate capacity, accessible for minor adjustments, able to withstand road shocks, and fundamentally simple. We have arrived at a design which very nearly satisfies all our requirements.

Fig. 8 shows the booster heater located on the engine side of a diversion valve. Water flows from the engine water manifold through a booster pump to the booster heater where heat is added at the rate of 75,000 Btu per hour when the heater is in operation. Flow is then to the heater core or directly back to the engine pump. The diversion valve is operated by the passenger space thermostat. When heat is required in the passenger space, water from the booster heater is shunted back to the engine by the diversion valve. Thus, the engine or coach, or both, can be heated by booster heater.

Fig. 9 is a space diagram of the units shown in Fig. 8. The booster heater is indicated by *4*. Water flow is from engine manifold *1* through heater water pump *2*, thence through booster heater *4* and diversion valve *3*. If thermostat *A* demands heat, diversion valve *3* directs water to heater core *5*. If thermostat *A* is satisfied, water flow is confined between engine and booster heater. When engine water temperature falls to a selected minimum, engine temperature switch *B* ignites booster heater and maintains operation until water temperature has reached desired maximum.

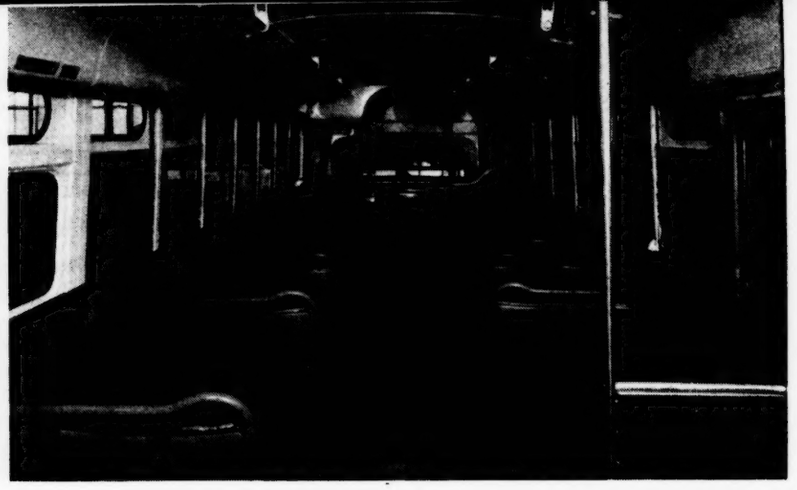


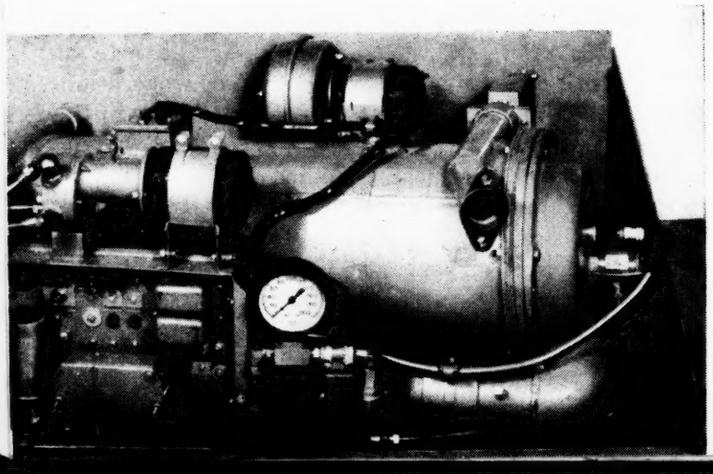
Fig. 11. Interior of GM coach with grille to booster heater visible at bottom of picture in front of side exit.

It is convenient at this point to discuss air flow by means of which coach temperature is maintained, odors are eliminated, and windows are kept clear. Air is drawn through the upper front roof panel by blower *6*, forced upward through heater core *5*, and into air ducts along both sides of coach above windows. Correct temperature of this air within the ducts is maintained by thermostat *A* which is located therein. Heated air is thus distributed and discharged through slots *8*. The direction of air discharge is downward over the windows. As previously shown, this air is dry or, in other words, has moisture absorbing capacity. In passing over the windows it removes or prevents moisture accumulation upon the glass.

Fig. 10 shows the booster heater package assembly which is approximately 28 inches long by 17 inches wide by 13 inches high.

Fig. 11 shows an interior view of a GM transit coach. In the lower left is seen the emergency exit directly opposite the rear door. Under the floor at this point is located the booster heater and just above it on top of the floor under the foreground seat is located the diversion valve and heater water pump. Water from the engine, after passing through the booster heater, flows to this diversion valve where it is directed forward to the front of the coach or shunted back to the engine, according to the passenger space thermostat requirements. Hot water flowing forward on demand of the thermostat circulates through a heater core directly over the driver's head and from there returns to the intake side of the engine water pump. Air is taken in directly over the front windshields, forced through the heater core, and along the ducts above the passenger windows on both sides of the coach. Just outboard of the two rows of lights, narrow slots are located. Air is discharged outwardly and curves down over the advertising card racks and finally flows over the windows and thence to the floor. The desired temperature of this air is maintained by periodic, rapidly intermittent flow of hot water as regulated by the diversion valve which, in turn, is operated by the passenger space thermostat. The thermostat is located in the front end of the coach, one element within and sensitive to duct temperature, and another element exposed and sensitive to passenger space temperature. The action of this thermostat is to increase temperature of air in the ducts as the passenger space temperature decreases, and, conversely, to decrease duct temperatures as passenger space temperature increases. In extremely cold weather duct temperatures are maintained relatively high with the result that the ducts which extend from directly above the grab rail to the windows give off an appreciable amount of radiant heat. The outwardly discharging air from the ducts produces an inflow of surrounding air from the ceiling with the result that the same temperature is maintained substantially from ceiling to floor, and from front to rear. The continuous flow of heated dry air over all windows prevents fogging and frosting.

Fig. 10. Janitrol booster heater package assembly.

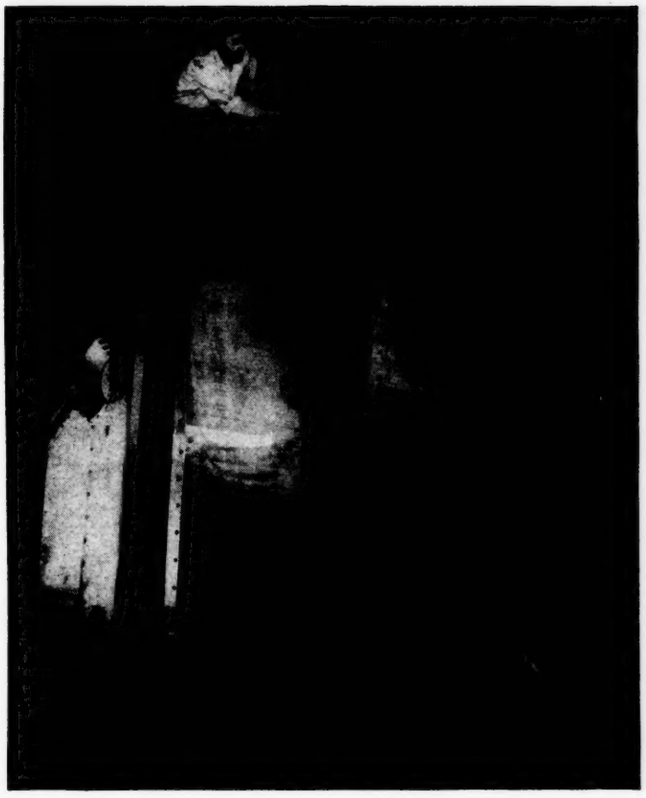


Although we have specified —30F as a design low temperature, it should be emphasized here that auxiliary heat is needed during the course of normal winter operation when outside temperature is below the comfort level. There are many reasons why auxiliary heat is desirable in addition to those already mentioned. The more important of these are: Prewarming of engine before attempting to start in cold weather with consequent low drain on the battery and elimination of crankcase dilution; reduced engine wear resulting from higher average operating temperatures; improved fuel economy resulting from maintenance of engine temperature near the normal operating level; outdoor winter parking with consequent reduced investment in plant equipment; and last but not least, healthier and more comfortable working conditions for the driver.

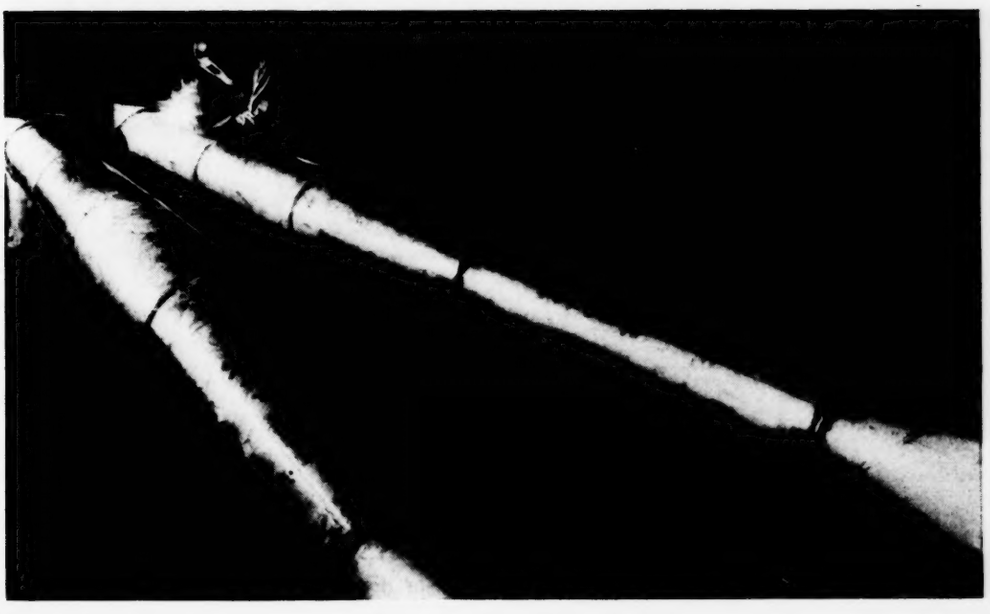
In conclusion, it may be definitely stated that the accomplished removal of odors, the maintenance of clear windows, and the even regulation and distribution of temperature throughout the coach would not be possible under many operating conditions without the addition of a fuel-burning booster heater.

Picture Paragraphs

(Above) A refrigerated, aluminum container, used in war to transport whole blood, now ships live lobsters and other perishables. Developed by George B. Wagner, left above, at a cost of $57,000, crate will be made by Allied Container Corp. Magnesium cooling unit in cover operates on gases from disintegrating dry ice, will run 12 days adjustable to either above or below freezing.



(Left) Housing for a big induced draft fan receives final adjustment at the B. F. Sturtevant Co. Division of Westinghouse Electric, Hyde Park, Mass. Fan will handle 90,000 cfm of hot gases in a southern paper company power plant.



(Right) These twin 80-ft exhaust stacks are made of 18.8 stainless steel sheet for exhausting acid fumes in a dye works. Steel sheets were first rolled and welded longitudinally then welded into stack sections using Airco 19-9 stainless steel electrodes. Photo courtesy Air Reduction Sales Co.

Reducing Condensate Temperature in Process Heating

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For cases where condensate returns at different temperatures in different lines, as frequently happens in process work, or where building heating and processes are both involved, the author shows various methods of employing heat exchangers to bring the condensate temperature to a point where the pump will be able to perform satisfactorily.

INTERESTING problems are often presented in the handling of condensate in returns where these are at different temperatures, at pressures insufficient for return to the boiler house by gravity, and at temperatures too high for a pump to handle. Such returns may be from heating and process mains, or process lines only, and those from process applications may fluctuate widely in quantity, if not in temperature. Returns may also be higher in both quantity and temperature than estimated, due to one or more defective traps blowing steam directly into the returns. One such trap when operating wide open may in itself upset all calculations.

A condensate pump is designed primarily to handle water, and hence has insufficient volumetric capacity for expanding vapors or steam which flash to a large volume from condensate at a higher temperature than the suction pressure prevailing at the pump. Normally, the pump should handle condensate at a temperature no higher than 200F, though by separating the receiver from the pump and elevating it a few feet to where the static head will provide sufficient pressure for water at correspondingly higher temperatures, condensate at a temperature of 215F or 220F may be successfully pumped.

Holding return line pressures as low as possible is a matter of good engineering. Aside from lessened heat transmission losses from return lines and therefore more comfortable spaces to work in during the non-heating season, the various steam traps will have less work to do when discharging into returns in which there are lower pressures. Traps will blow less steam into the returns at such times. Partially offsetting such advantages is the greater amount of flash steam from condensate when emptied into low pressure returns. Where much flash can be utilized, there is, however, no disadvantage, since such utilization will keep the pressures down, conserve heat, and improve the effectiveness of the process.

Before taking up the question of detailed arrangement, it is important that the scheduling or timing of the process be thoroughly analyzed so that condensate quantities will in actual practice reasonably approximate those as calculated. For a number of reasons, the poundage of condensate per hour may not be the same as the pounds of steam per hour upon which the steam mains were sized. For example, steam may go directly into the batch or process for bringing the process material and its container up to temperature and therefore will not be returned. In addition, not all equipment is on at the same time, and while the maximum number of like equipment on at a time may be estimated, this number will vary in location at times. It may be the first or the last of such equipment sup-

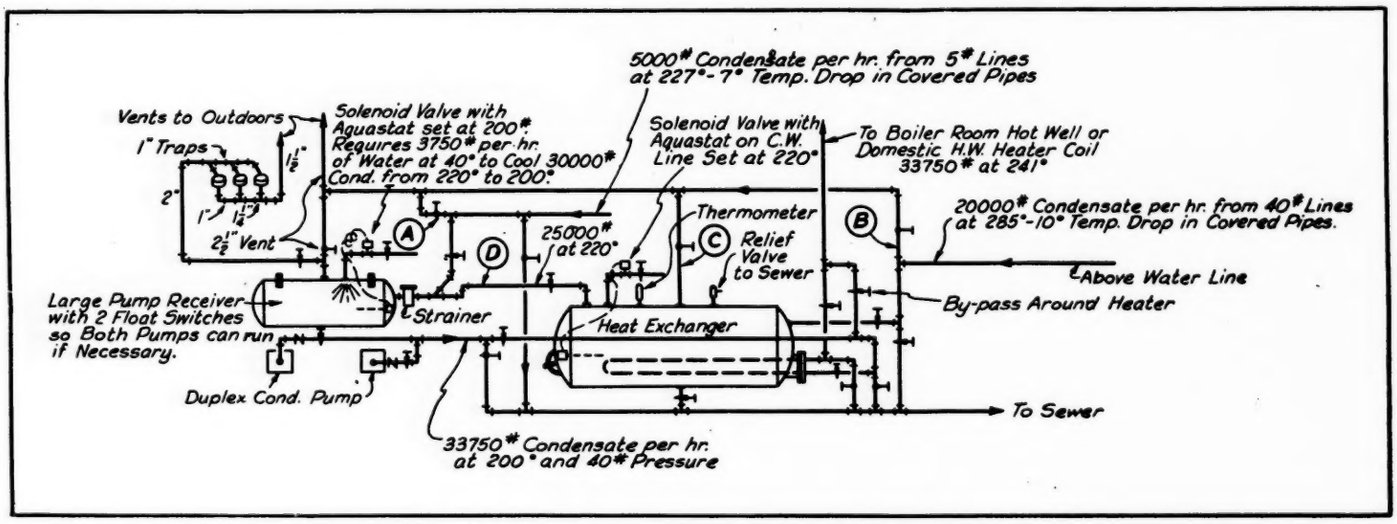
5000# Condensate per hr. from 5# Lines at 227°-7° Temp. Drop in Covered Pipes
Vents to Outdoors
1" Traps
Solenoid Valve with Aquastat set at 200#. Requires 3750# per hr. of Water at 40° to Cool 30000# Cond. from 220° to 200°.
Solenoid Valve with Aquastat on C.W. Line Set at 220°
To Boiler Room Hot Well or Domestic H.W. Heater Coil 33750# at 241°
2"
1½"
1¼"
2½" Vent
A
D
25000# at 220°
Thermometer
C
Relief Valve to Sewer
B
20000# Condensate per hr. from 40# Lines at 285°-10° Temp. Drop in Covered Pipes.
Above Water Line
By-pass Around Heater
Large Pump Receiver with 2 Float Switches so Both Pumps can run if Necessary.
Strainer
Heat Exchanger
Duplex Cond. Pump
To Sewer
33750# Condensate per hr. at 200° and 40# Pressure

Fig. 1. Diagrammatic arrangement of heat exchanger and pump using jet water at the pump.

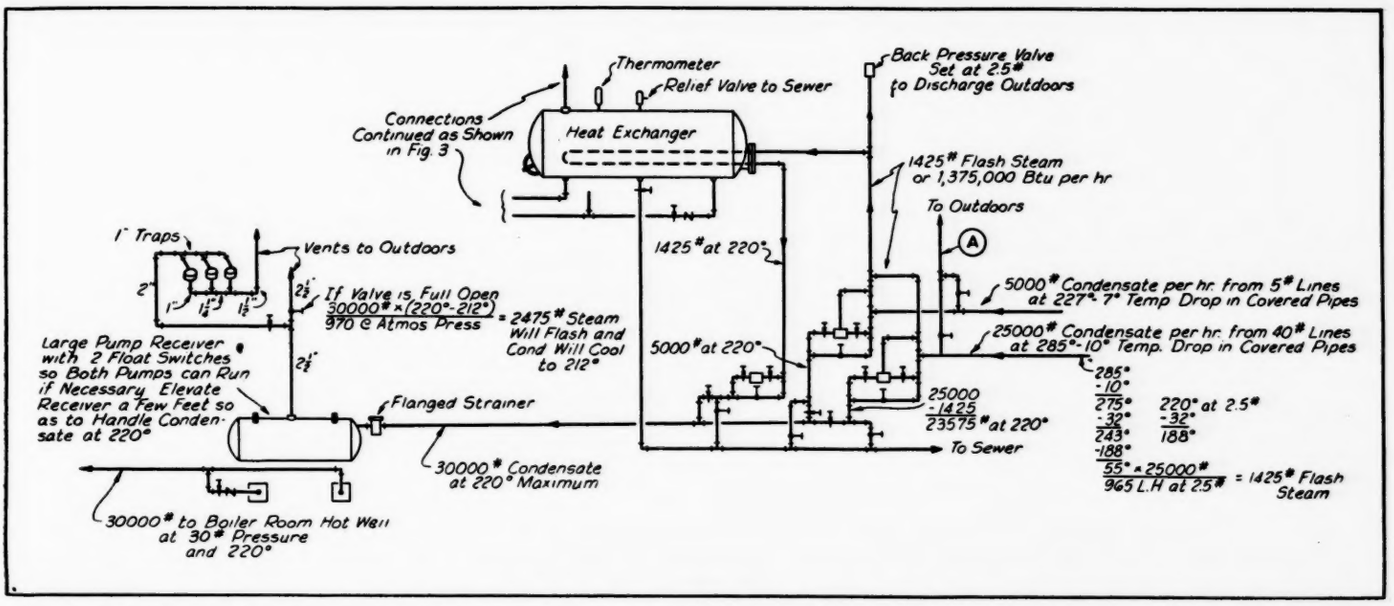
Thermometer
Relief Valve to Sewer
Back Pressure Valve Set at 2.5# to Discharge Outdoors
Connections Continued as Shown in Fig. 3
Heat Exchanger
1425# Flash Steam or 1,375,000 Btu per hr
To Outdoors
1" Traps
Vents to Outdoors
1425# at 220°
5000# Condensate per hr. from 5# Lines at 227°. 7° Temp. Drop in Covered Pipes
25000# Condensate per hr. from 40# Lines at 285°-10° Temp. Drop in Covered Pipes
If Valve is Full Open 30000# x (220°-212°) / 970 @ Atmos Press = 2475# Steam Will Flash and Cond Will Cool to 212°
5000# at 220°
Large Pump Receiver with 2 Float Switches so Both Pumps can Run if Necessary. Elevate Receiver a Few Feet so as to Handle Condensate at 220°
Flanged Strainer
25000 - 1425 / 23575# at 220°
To Sewer
285° - 10° / 275° - 32° / 243° - 188° / 55° = 25000# / 965 L.H at 2.5# = 1425# Flash Steam
220° at 2.5# - 32° 188°
30000# Condensate at 220° Maximum
30000# to Boiler Room Hot Well at 30# Pressure and 220°

Fig. 2. Diagrammatic arrangement of heat exchanger or preheater and domestic hot water heater utilizing flash steam of the condensate.

plied, yet in each case the steam main must be large enough at the point of its run where the particular equipment is supplied, whether first or last.

Still other factors must be considered, for while heating loads are practically a steady flow per hour, a certain process may require steam for only a fraction of an hour—for example, ten minutes. In such a case, the process supply and return mains when sized on the hourly basis, should be sufficient in size for a load six times as great as the condensate produced in the ten minute interval. In this example, 1000 lb of steam in ten minutes calls for a main large enough to handle 6000 lb of steam per hour, but the heat available at the heat exchanger for storage and heat balance against heat requirements of a hot water heater is only that as obtained from 1000 lb of condensate per hour. All this may be obvious, particularly so when pointed out, but experience has proved it not so obvious in view of present day confusion. The point is, do not take the pounds of steam set down for the sizing of the mains as the pounds of condensate estimated per hour for heat recovery. In the final analysis include also the condensation as returned from steam main drips, which may be 25% in the case of heating mains and 5 to 10% for process mains.

Discussing the various methods of accomplishing the desired results, consider in Fig. 1, a most simple arrangement, satisfactory in every way. It is assumed that the additional jet water can be used as make-up water to the boilers and that the untreated jet water is such a small percentage of make-up requirements as not to affect any make-up already treated by water softeners.

The novel feature of this scheme is in the utilization of the pump discharge through a heating or cooling coil (whichever way you look at it) in the heat exchanger so as to cool the condensate down to where it can be handled by the pump while returning the heat as reclaimed to the hot well in the boiler room. The discharge of the pump being under pressure, the temperature of the discharge can go as high as 285F (the temperature corresponding to 40 lb pressure) and still not flash. The discharge is at 40 lb due partly to the coil resistance. Note that the hot water discharge from the heat exchanger could be passed through an additional coil (as in a domestic hot water heater) on its way to the hot well.

It is also interesting to note in Fig. 1 that the returns from heating lines at 5 lb pressure come back after heat transmission losses from insulated returns at a temperature of 220F. This fixes the setting of the solenoid valve on the heat exchanger to maintain a corresponding temperature of water leaving the heat exchanger should the condensate pumped through the coil fail to do so in itself. Obviously, if the temperature drop in covered heating returns is 7F, as noted in Fig. 1, and the efficiency of the covering is 75%, then the temperature drop in bare returns would have been 28F, which subtracted from 227F (the temperature at 5 lb pressure) is about 200F.

Such a temperature would have eliminated the need for any jet water, but would have required a fan of about 31,000 cfm (based on a 10F temperature rise) to remove some of the heat back in the working spaces on a very hot day. Leaving the process mains bare, as sometimes proposed, would require, in the given case, an additional fan of about 26,000 cfm to provide some comfort. Such an investment in fan equipment and the cost for power consumption would not be warranted, even without consideration of the heat lost and the discomfort entailed. Should the space be air conditioned, the mains should most certainly be covered and with double thickness of high grade insulation.

The returns in Fig. 1 must be kept above the water level within the pump receiver. The coil in the heat exchanger should be submerged and the lower the heat exchanger is set the less likely there is of steam pressure becoming high enough suddenly to lift and empty the heat exchanger contents over into the pump receiver. As a hedge against difficulties, such as would be caused by a flooded receiver, a number of precautionary measures have been taken, including an oversize receiver and two float switches set at different levels with a manual throw over switch, so that the

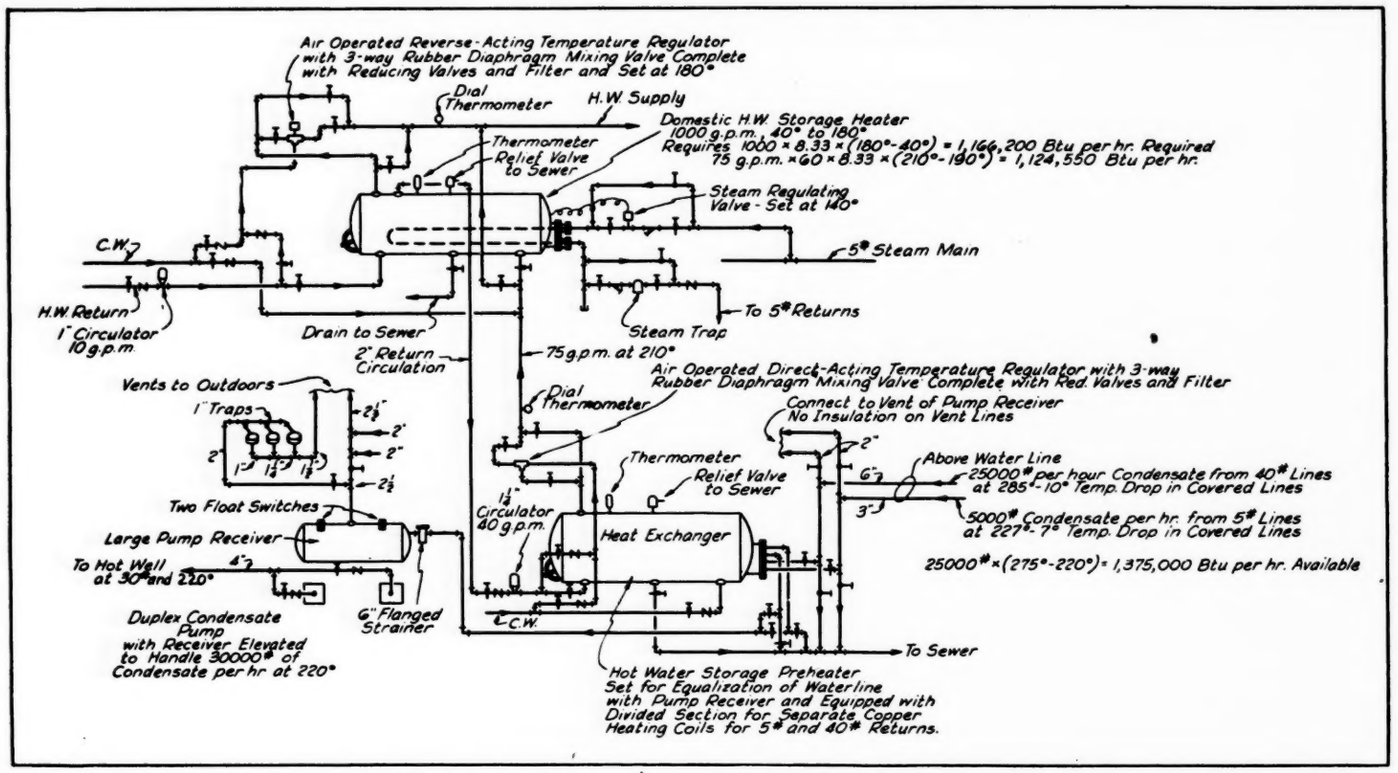
Air Operated Reverse-Acting Temperature Regulator with 3-way Rubber Diaphragm Mixing Valve Complete with Reducing Valves and Filter and Set at 180°
Dial Thermometer
H.W. Supply
Domestic H.W. Storage Heater
1000 g.p.m. 40° to 180°
Requires 1000 × 8.33 × (180°-40°) = 1,166,200 Btu per hr. Required
75 g.p.m. × 60 × 8.33 × (210°-190°) = 1,124,550 Btu per hr.
Thermometer
Relief Valve to Sewer
Steam Regulating Valve - Set at 140°
C.W.
5# Steam Main
H.W. Return
1" Circulator 10 g.p.m.
Drain to Sewer
To 5# Returns
Steam Trap
2" Return Circulation
75 g.p.m. at 210°
Air Operated Direct-Acting Temperature Regulator with 3-way Rubber Diaphragm Mixing Valve Complete with Red. Valves and Filter
Vents to Outdoors
1" Traps
Dial Thermometer
Connect to Vent of Pump Receiver No Insulation on Vent Lines
Thermometer
Relief Valve to Sewer
Above Water Line
25000# per hour Condensate from 40# Lines at 285°-10° Temp. Drop in Covered Lines
5000# Condensate per hr. from 5# Lines at 227°-7° Temp. Drop in Covered Lines
Two Float Switches
1½" Circulator 40 g.p.m.
Heat Exchanger
Large Pump Receiver
To Hot Well at 30° and 220°
25000# × (275°-220°) = 1,375,000 Btu per hr. Available
Duplex Condensate Pump with Receiver Elevated to Handle 30000# of Condensate per hr. at 220°
6" Flanged Strainer
C.W.
To Sewer
Hot Water Storage Preheater Set for Equalization of Waterline with Pump Receiver and Equipped with Divided Section for Separate Copper Heating Coils for 5# and 40# Returns.

Fig. 3. Diagrammatic arrangement of heat exchanger or preheater and domestic hot water heater using direct cooling of the condensate.

standby pump will operate as an auxiliary on peak loads.

To prevent air binding before condensate drops below the water level there is a vent line at *B*. As for the condensate line at *A*, it can vent air through the vent on the pump receiver, or direct as shown. Lines at *A*, *B*, and *C* serve as vents to prevent steam locking and are fitted with globe valves which can be throttled so as to hold back some pressure and not waste too much flash steam out through the main vent. Condensate line at *B* and line *C* could be arranged with steam traps to vent air as shown at the pump receiver and so as to keep some pressure on the lines to prevent unnecessary flashing.

The thermostatic traps shown are employed to obtain the necessary capacity with this type of trap. In lieu of the traps, the globe valve on the receiver vent may be throttled and yet pass enough steam and air. The traps do allow a limited pressure in the tank which causes the pump to work better when handling very hot condensate. Incidentally, a pump having a low number of revolutions is preferable when the condensate is extremely hot.

Jet water is sprayed and its flow is controlled by an immersion aquastat connected to a solenoid valve, so as to operate only when absolutely necessary. The setting of the aquastat at the heat exchanger is at 220F, but a higher setting may be found necessary in some cases, so as to prevent cooling of the condensate prematurely before the cooling coil has full opportunity to do its work. Condensate returns are also arranged, so that either one may go (all or in part by throttled globe valve) direct to the sewer. The pump discharge may do likewise, either around the heat exchanger or after cooling in it through the coil. This discharge may also by-pass the heat exchanger so as to go direct to the hot well.

A further comment is that if desired the tank can be raised slightly, in which case a high pressure trap could be installed at the point *D* so as to prevent the passage of steam and yet permit the passage of air and condensate at the temperature for which the trap is set. The quantity of condensate in the given case is so great that more than one trap would be required. As to the pump size, it should be based on a gallonage of water twice that of the condensate returned from the process and three times that of the condensate returned from the heating mains. Condensate from process may at times come in surges and present some problems in overburden and overheating, but the first heating up of cold heating mains and equipment is a greater load.

This brings us to a consideration of the method shown in Fig. 2 wherein steam is flashed from the condensate down to 220F (the setting of the back pressure valve) for utilization in the heat exchanger. The difference is sensible heat at the temperatures returned and that corresponding to the setting of the back pressure valve, when divided by the latent heat at the lower pressure, gives the pounds of steam flashed to the coil in the heat exchanger. The water heated by the coil then goes to additional storage as provided by a domestic hot water heater in the same manner as arranged in Fig. 3.

Traps and returns are fitted with by-passes and a by-pass is provided at *A* to permit dropping of the pressure to atmospheric, should this be desired at any time. Air is relieved from and before the float traps to the back pressure valve when this valve is open and at other times through the coil and its combination float and thermostatic traps to the vented pump receiver. More than one trap may be required for the 40 lb line in order to obtain capacity. If the valve on

the pump receiver is opened wide steam will be lost due to flash down to 212F at atmospheric pressure.

In Fig. 3, the condensate is passed through two separate coils in a divided section of the heat exchanger, one coil for each return. Condensate enters the bottom of each coil and leaves at its top, thus keeping the coils full of condensate. The heat exchanger can, if necessary, be elevated above the pump receiver, though in such a position the coil can more easily be blown free of condensate by pressure in the returns. The returns must be kept above the water line of the pump receiver so as to prevent backflow of condensate and for venting of these lines of steam and air to the vent of the pump receiver. Equipment such as tanks, traps, and mixing and regulating valves is by-passed by condensate, steam and cold water lines, and certain lines are equipped with globe valves as in Fig. 1 and 2. Check valves prevent backward flow as shown. Venting must be from the top of the returns just before they drop to submerge below the water line. Vent lines should be left without insulation so that they may provide sufficient surface to condense steam when not in too great a quantity and thus save this steam from being wasted through the vent of the pump receiver.

Note that the heat available from the condensate returns is slightly more (as per calculations on the sketch) than the heat requirements of the domestic hot water heater. Both heater and heat exchanger are of the storage type to store heat on surges of condensate. When there is insufficient condensate for heating the water the steam regulating valve on the domestic hot water heaters cuts in so as to maintain a temperature of at least 140F. If this valve were set at 180F it would at times pass steam so as to heat the water circulated to the heater before this water had an opportunity to become fully heated by the heat exchanger.

The hot water heater is provided in the usual way with a hot water return and a small circulator. A second circulator, one large enough to keep the two tanks at somewhere near the same temperature, is required when the upper tank is some distance away from the lower tank and not far enough above it to provide sufficient circulation by gravity. Each hot water return should be connected to its respective mixing valve, so that the circulator can still pass its water whenever the mixing valve throttles against flow through it of overheated water from the tank. The mixing valve at the heat exchanger is not absolutely necessary, but it does serve to prevent an overheated hot water heater and comes in handy when the hot water heater is temporarily out of service for the cleaning of its coils, or should the upper mixing valve fail.

The lower mixing valve is direct acting while the upper one is reverse acting. Should the air supply fail to the lower valve, the valve springs will move the disc so that the cold water supply will be shut off and only hot water will be passed through the mixing valve. This insures continued heating. Should the air supply fail to the upper valve, the valve springs will move the disc so that only cold water will be passed through it. This is the usual safety precaution against dangerously hot water at plumbing fixtures.

Low Temperature Research

Interest in low temperature research, which has increased greatly in recent years, promises to be even more widespread through the recent development by Professor S. C. Collins of Massachusetts Institute of Technology of a relatively simple apparatus known as the helium crysostat.

It makes readily available any temperature down to — 457F. The helium crysostat operates by circulating gaseous helium at moderate pressure through a heat exchanger and then expanding it through an engine where it does the work. Consequently more heat is removed from the helium than if it were expanded in the conventional manner through an orifice.

Thermodynamically, this engine is the heart of the process and because of its simplicity and effectiveness, represents a remarkable achievement. It must handle what is one of the thinnest fluids with high efficiency, but without lubrication and without friction, for at such low temperatures lubricants would solidify and ordinary steel becomes brittle.

By constantly circulating helium and cooling it in the heat exchanger with the cooler helium from the engine exhaust, any heat leakage due to imperfect insulation may be counteracted.

As contrasted with the tons of special equipment operated at pressures up to 3,000 lb per sq in. to produce liquid helium by some of the other methods now in use, the present device weighs approximately one-half ton, three-quarters of which is devoted to the compressor and drive. It operates at only 200 lb per sq in.

The new crysostat is expected to find important use in metallurgical research on thermal conductivity, embrittlement, and many other characteristics of metals and alloys. Present metals attain only a small fraction of the strength which theoretical consideration of the forces between and within molecules indicate to be possible. The magnetic interaction between protons and neutrons within the nucelus of an atom is also one subject which can be studied best at low temperatures.

Research in the low temperatures has heretofore been limited mainly to certain fixed temperatures, such as those of liquid oxygen, nitrogen, hydrogen or neon. The new device, however, can be operated so that its generation of cold just balances heat leakage into it or heat generated by experimentation, and any desired temperature may be obtained.

Graphical Determination of Heat Transfer Film Coefficient for Steam

B. F. RABER* and F. W. HUTCHINSON**

THE two graphical solutions presented in this article are for superheated water vapor, steam, being either heated or cooled (but not condensed) in the temperature range from 50F to 500F. These solutions supplement the two which have already been presented (H&V, December, 1944) for the heating or cooling (but not evaporation) of water. Unlike most of the liquids which have been treated in preceding articles, steam inside a pipe has the same film coefficient whether heating or cooling; hence only one set of temperature lines need be given.

One very significant difference exists between the graphs for steam and those which were given for liquids. In the case of all liquids (Fig. 1 through 20) the fluid velocity, V, in feet per second, has been plotted as one of the variables of the system. For steam (and for subsequent solutions for other vapors) a weight rate of flow, W, in pounds per second per square foot of cross-section has been used in place of velocity; then

$$W = wV$$

where: w = specific weight of the vapor in pounds per cubic foot

V = flow velocity, feet per second

Fig. 21 is for turbulent flow of superheated steam within a pipe whereas Fig. 22 is for flow outside and normal to a single pipe. Fig. 21 is valid for all pressures lower than 50 lb per sq in. abs., but for 100 lb per sq in. abs. it is not valid if W is less than 1½ and for 150 lb per sq in. abs. it is invalid if W is less than 2.0. The dashed example line on Fig. 21 shows that for a flow rate of 33 lb per sec per sq ft through a 2½-in. ID pipe and at a temperature of 300F the film coefficient, heating or cooling, would be 76 Btu per (hr) (sq ft) (F). Fig. 22 gives a dashed example line showing h_o of 36 for flow outside a 2½-in. OD pipe at W of 13 and temperature of 250F.

Evaluation of Approximate Overall Coefficient

Preceding articles have given tables of resistance for use in calculating the overall coefficient of heat transfer per lineal foot of various standard sizes of pipe and tube. As an aid in calculations for non-standard pipe and tube and for use in achieving rapid approximation to the overall coefficient for any pipe or tube, Table 6 gives the reciprocal of the total resistance represented by any combination of values of the inside and outside film coefficients. Since the resistance of metal pipe walls is extremely low with respect to the resistance of the two films, it follows that the reciprocal, H, of the total film resistance gives a close approximation to the overall coefficient for most systems. Then,

$$H = \frac{1}{[(1/h_i) + (1/h_o)]}$$

$$= \frac{1}{[(1/h_i) + (L/k) + (1/h_o)]} = U$$

When non-metal pipes are used, or when very high flow velocities give values of h_i and h_o greater than 500, it will usually be desirable to calculate U; in all other cases H can be used with accuracy sufficient for most practical purposes.

*Professor of Mechanical Engineering, University of California, Berkeley, Cal.

**Professor of Mechanical Engineering, Purdue University, Lafayette, Indiana.

Table 6 — Values of the Reciprocal, H, of the Total Film Resistances

Inside Film Coefficient, h_i	Outside Film Coefficient, h_o									
	100	200	300	400	500	600	700	800	900	1000
100	50	67	75	80	84	86	88	89	90	91
200	67	100	120	133	143	150	155	160	164	167
300	75	120	150	172	188	200	210	218	225	231
400	80	133	172	200	222	240	254	267	277	286
500	84	143	188	222	250	273	292	308	322	333
600	86	150	200	240	273	300	323	343	360	375
700	87	155	210	254	292	323	350	373	394	411
800	89	160	218	267	308	343	373	400	424	445
900	90	164	225	277	322	360	394	424	450	474
1000	91	167	231	286	333	375	411	445	474	500

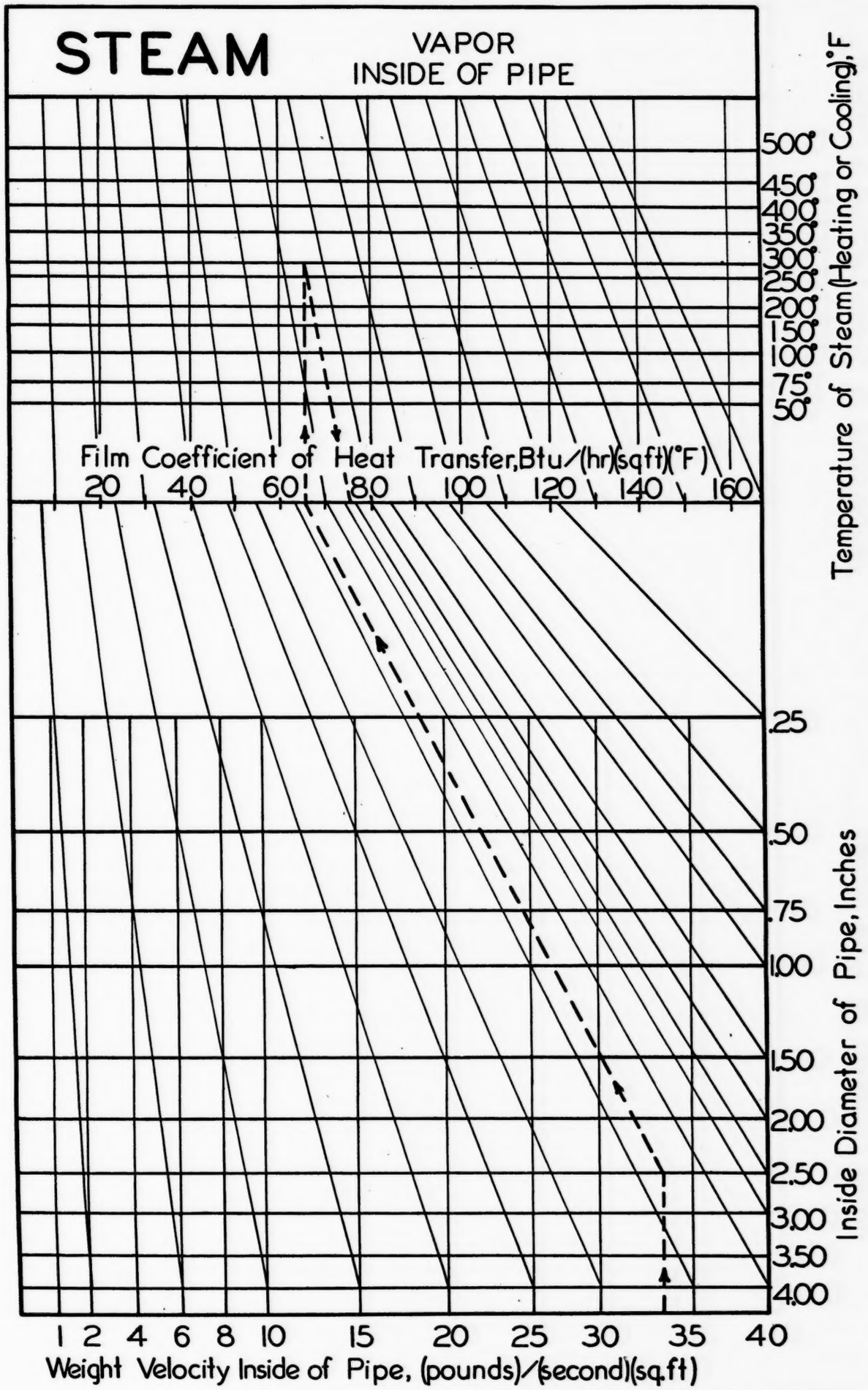
STEAM
VAPOR
INSIDE OF PIPE
Film Coefficient of Heat Transfer, Btu/(hr)(sq ft)(°F)
20 40 60 80 100 120 140 160
Temperature of Steam (Heating or Cooling), °F
500° 450° 400° 350° 300° 250° 200° 150° 100° 75° 50°
Inside Diameter of Pipe, Inches
.25 .50 .75 1.00 1.50 2.00 2.50 3.00 3.50 4.00
1 2 4 6 8 10 15 20 25 30 35 40
Weight Velocity Inside of Pipe, (pounds)/(second)(sq.ft)

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Fig. 21. Film coefficient for steam flowing inside a pipe.

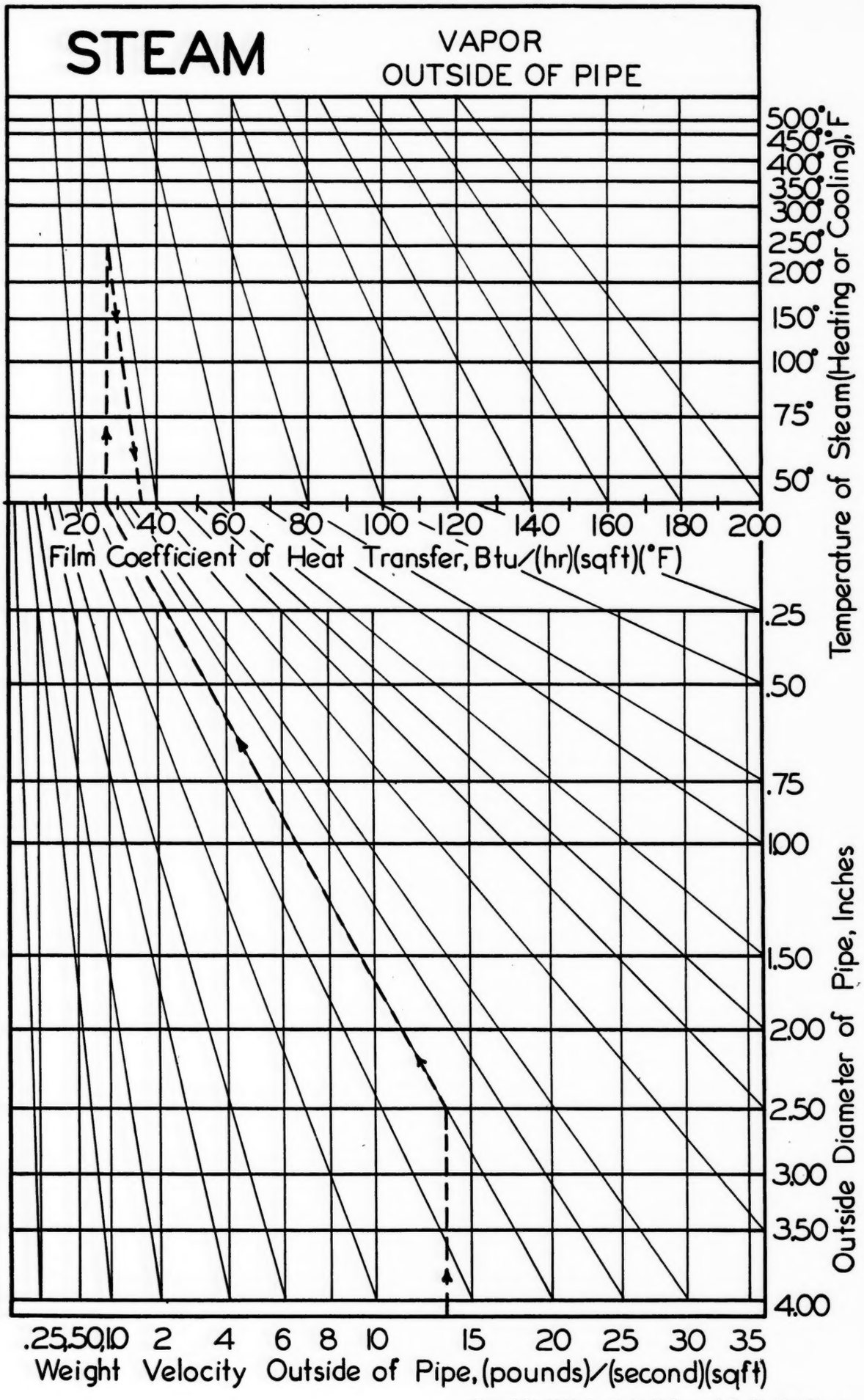

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Fig. 22. Film coefficient for steam flowing outside a pipe.

NEWS OF EQUIPMENT AND MATERIALS

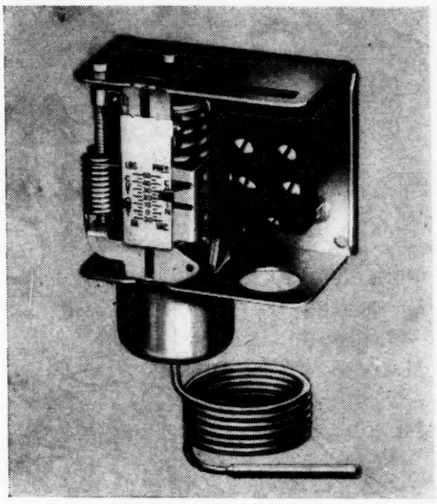
Diaphragm Gas Valve

NAME — Minneapolis - Honeywell diaphragm gas valve and pressure regulator.
PURPOSE — For providing pressure regulation on gas line.
FEATURES—Controller is of the relay-operated type, and operating parts are placed out of the gas circuit. Spring-type pressure regulator is said to provide good regulation under varying line pressures and has a convenient adjustment means for varying outlet pressures. Adjustable bleed restrictor permits slow opening delay.
SIZES AND CAPACITIES — ½ in. for 250 cfh; ¾ in. for 400 cfh; 1 in.

for 600 cfh. Pressure rating, 8 oz per sq in.
MADE BY—*Minneapolis - Honeywell Regulator Company, Minneapolis, Minn.* **222**

Refrigeration Control

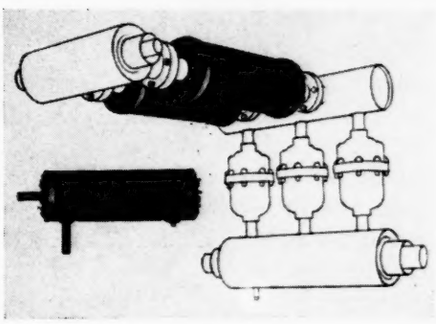
NAME—Penn 270 refrigeration control.
PURPOSE—Switch control based on refrigeration operation.
FEATURES—Incorporates two separate circuits for the control of single phase multiple systems and polyphase systems where one pole is used as a pilot control. This one control can start simultaneously with the condensing unit and open the magnetic valve. Unit incorporates a simple calibrated scale which



shows directly both cut-in and cut-out points. This makes it easy for the operator to set the control points. Unit controls two separate load circuits, controls multiple systems, controls polyphase motors without use of magnetic contractors or line starters, and is said to provide greater protection by breaking both lines of current supply.
LITERATURE AVAILABLE—Illustrated folder.
MADE BY — *Penn Electric Switch Co., Goshen, Ind.* **223**

Heat Transfer Drier

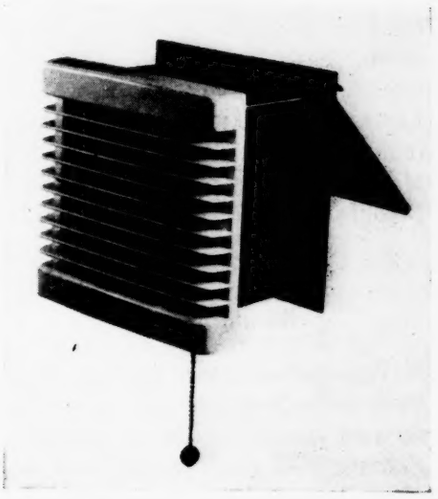
NAME—Goodyer Aridizer.
PURPOSE — Dehumidifying compressed air.
FEATURES—The heating of compressed air for drying and dehumidifying is made possible by



the bilateral fin design of the unit. Temperature of the compressed air for a dehydrating process can thus be considerably lower than if the dehydrating were accomplished by a conventional space heater type drier. Unit is small and compact with accessibility to all vital parts. Heating may be effected by electric calrods, oil or gas burning equipment. Unit can be constructed as an integral part of compressed air, gas or steam system. Temperatures up to 1000F, it is claimed, can be maintained with control of leaving temperatures to within plus or minus 1F.
LITERATURE AVAILABLE — Bulletin 14.
MADE BY—*Goodyer Industries, Inc., 224 S. Michigan Ave., Chicago 4, Ill.* **224**

Home Ventilator

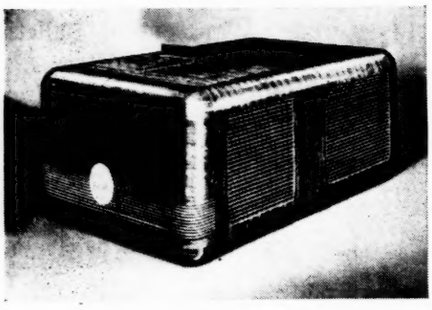
NAME—Aeropel home ventilator.
PURPOSE — For improving ventilation in rooms in homes.
FEATURES—Unit is constructed with



a white plastic grille which can be attached with the bars in either horizontal or vertical position. The tube-axial vane consists of 3-bladed aerocurve propeller driven by an enclosed 115, 60 or 50 cycle, single phase motor. A resilient spring mounting attaches the motor to the cylindrical air tunnel. An automatic latching mechanism is furnished with each unit and the fan starts automatically when the door is opened. Ventilator has a capacity of 400 cfm.
MADE BY—*American Blower Corp., Detroit, Mich.* **225**

Air Conditioner

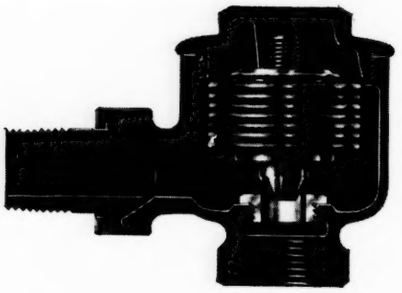
NAME — Pacific air conditioner, model 50WA.
PURPOSE—Room air conditioner.
FEATURES — Unit is said to have capacity sufficient to cool a room 14 by 16 ft with an average ceiling height of 9 ft. Cabinet is lined with 1 in. thick Fiberglas for insulation against noise and heat transmission. Unit has a twin cylinder vertical type open compressor



with twin V-belts to compressor motor. A fin type copper coil evaporator is used. Refrigeration capacity of unit is 5,000 Btu per hr. Cabinet is available in four types of finishes. It can be set in any window more than 22¼ in. wide.
LITERATURE AVAILABLE—Illustrated folder.
MADE BY—*Pacific Mfg. Corp., 5308 Blanche Ave., Cleveland 4, O.* **226**

Radiator Trap

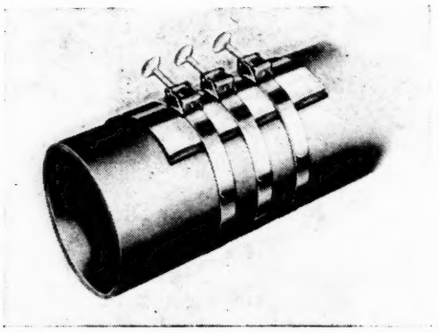
NAME—Low-pressure radiator trap.
PURPOSE — For use in vapor and vacuum steam heating systems.
FEATURES—Valve opening is effected by a slight temperature difference between steam and condensate. The trap is of the thermostatic bellows type based on the balanced vapor-pressure principle. Bellows is of bronze, formed from seamless tubing. This low-pressure unit features a large diameter valve orifice for maximum water and air discharging capacity. The nickel alloy valve is welded integrally to the bellows to assure alignment. Trap has a renewable stainless steel seat. Body, cap and union are of corrosion resistant brass.
SIZES AND CAPACITIES—Provided in two types for 200 and 400 sq ft of



equivalent direct radiation.
LITERATURE AVAILABLE — Bulletin 744.
MADE BY—*W. H. Nicholson & Co., 199 Oregon Street, Wilkes-Barre, Pa.* **227**

Pipe-Leak Clamp

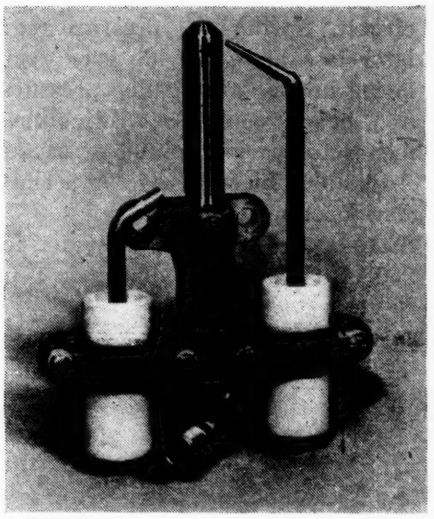
NAME — Marman stainless steel pipe-leak clamp.
PURPOSE—For eliminating leaks in pipes where pressure does not exceed 240 lb per sq in.
FEATURES—Clamp consists of a pad resistant to deterioration by water, gas, oil, or gasoline. An annealed backing plate covers the pad and



can be formed to the desired pipe contour. Pad and patch plate are secured over lead with stainless steel clamps. It is possible to use one of these clamps for several pipe diameters.
SIZES AND CAPACITIES — Clamps to fit pipe diameters from ½ in. to 12 in.
MADE BY—*Marman Products Co., Inc., 940 W. Redondo Blvd., Inglewood, Calif.* **228**

Pilot Burner

NAME — Minneapolis - Honeywell pilot burner for use with R177B electronic protectorelay.
PURPOSE — For providing an ignition and flame detection unit adapted to existing gas burner equipment.
FEATURES—Pilot burner is equipped with an adjustable primary air shutter and fixed orifices for manufactured or natural gases. It has one ignition electrode and one flame detection electrode. Burner tips are



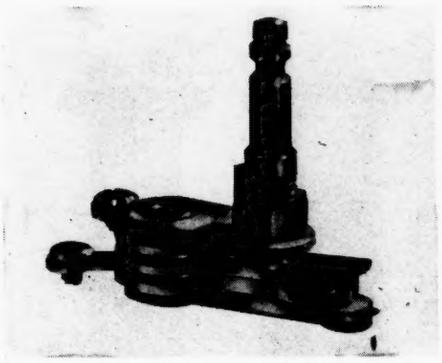
available for vertical or horizontal mounting. Unit has a tip with port straight off end, and one with twin ports 90 deg. to the center line of the tube and in a plane 90 deg. to the plane of the mounting bracket.
MADE BY — *Minneapolis-Honeywell Regulator Company, Minneapolis, Minn.* **229**

Rivet Gun

NAME—Cherry rivet G-55 hand gun.
PURPOSE—For use with small quantities of the various types of cherry rivets.
FEATURES—Tool was designed especially for small-quantity users of Cherry blind rivets in sizes of 7/32 in., ¼ in., and 9/32 in. When used with an adapter, it can also install smaller rivets. Hand gun can be used where air pressures are not available. It operates on ratchet principle, is 29 in. long including the pulling head, and weighs 4½ lb complete.
MADE BY — *Cherry Rivet Co., 231 Winston Street, Los Angeles 13, Calif.* **230**

Thermostat

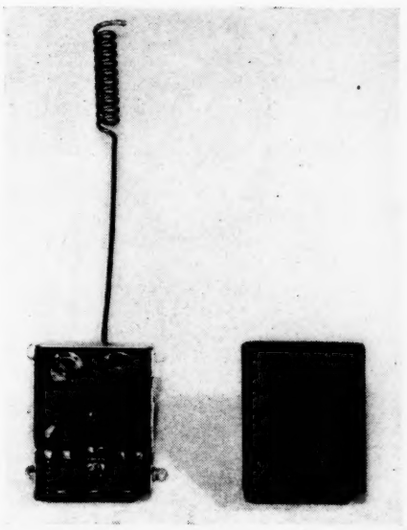
NAME—Stevens type S thermostat.
PURPOSE—Temperature control.
FEATURES — An independent bi-metal strip actuates the stainless steel or Inconel contact springs, and since the bi-metal carries no current, induction heating of the element does not take place. Heavy duty silver contacts are spot welded to the contact springs. Units can be supplied with either lava or mica insulation, and with either standard or elevated terminals.
SIZES AND CAPACITIES — Heating capacity listed as 15 amperes, 115



volts a-c. Available in a variety of lengths and styles.
MADE BY—*Stevens Manufacturing Co., Mansfield, Ohio.* 231

Furnace Control

NAME — Minneapolis - Honeywell combination fan and limit control.
PURPOSE—A control and switching device for preventing the fan control from being set above the limit control setting.
FEATURES—Flexible element is 20 in. long, or as an available special, there is a 40 in. long element. About 5 in. of this is coiled to provide the maximum surface area exposed to temperature being controlled. This makes the device very sensitive to temperature changes. A finger tip dial setting of the limit control is provided over a range of 170-250F for burner "off" point. The differential is 25F at midscale. Fan control has separate on and off finger tip dial adjustment with a range of 90-150F, with a minimum differential setting of



15F. It has a manual fan switch for summer operation. Maximum ambient temperature of control is 200F.
AVAILABILITY—Spring, 1948.
MADE BY — *Minneapolis-Honeywell Regulator Company, Minneapolis, Minn.* 232

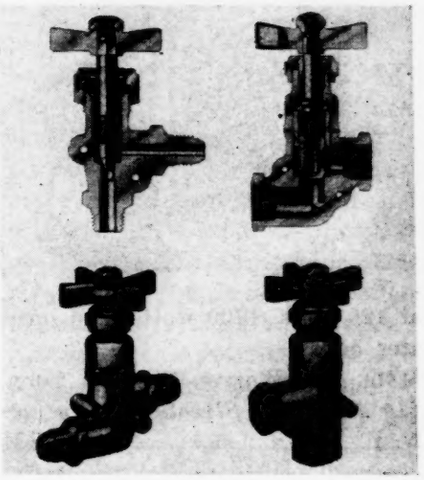
Defrost Control

NAME — White-Rodgers automatic defrost control, Type 2413.
PURPOSE—A combined control for refrigeration temperature and automatic defrosting.
FEATURES — Temperature control,

which has a range of —20 to 50F, is actuated by hydraulic action to give positive accuracy and ease of adjustment. Unit has a 7 ft capillary with special 15½ in. air sensitive bulb. Control contains a timer that has a clock and a 24-hour dial. Timer is adjustable from 15 minutes to 5 hours in 15-minute steps. Pointers on dial show the length of time that the defrost control is on and the compressor circuit is open. Five terminals on the control panel permit an arrangement whereby the defrost heater and the clock operate on 115 volts while the compressor circuit may be operated at 220 volts. Defrost cycle is usually set for some time during the night at which time the circuit to the compressor is opened and that to the defrost heaters closed.
MADE BY—*White-Rodgers Electric Co., 1209 Cass Ave., St. Louis 6, Mo.* 233

Industrial Valve

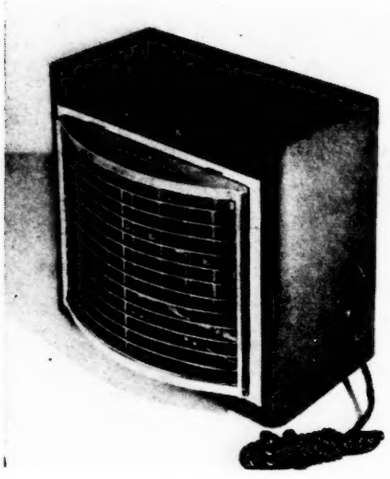
NAME—Parker valves.
PURPOSE—To provide an industrial valve of improved design.



FEATURES—Valves are available in needle and globe types and in six variations of connection arrangements. Straightline types are produced with female pipe threads or tubing threads. Angle types are provided with female pipe threads, with tube threads, or with male pipe threads at the inlet connection and either tube or female pipe threads at the side outlet. Valves have integral mounting lugs for convenience in securing valves to brackets or panels. Bolt holes are drilled to accommodate standard size bolts. Shut-off valves are of the non-rising stem type and can be repacked under full-flow or no-flow conditions.
MADE BY — *Parker Appliance Co., 17325 Euclid Avenue, Cleveland, Ohio*.....234

Heater Ventilator

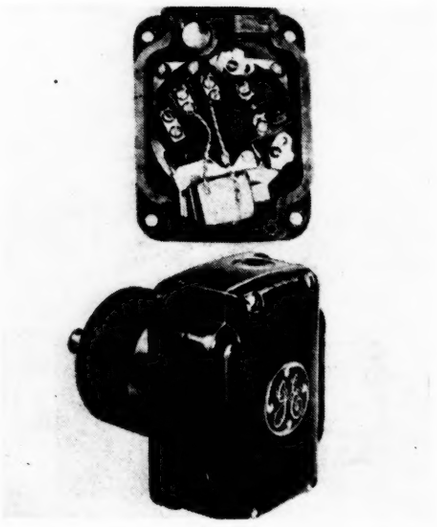
NAME—Air-D-Lux.
PURPOSE—An all-electric unit providing combination of heating-cooling-cooking facilities for the home.
FEATURES—A motor driven fan, controlled for various speeds, sends air across joined element-heated reflector fins and into the room. When cooling unit is wanted, the heat element can be switched off. By placing the unit on its back and raising the hinged grille a cooking unit is provided. Heater is rated



at 120 volts, 1700 watts, and operates on a-c.
MADE BY—*Warren Simpson Corp., 110 East 3rd Street, Mt. Vernon, N. Y.* **235**

Plugging Switch

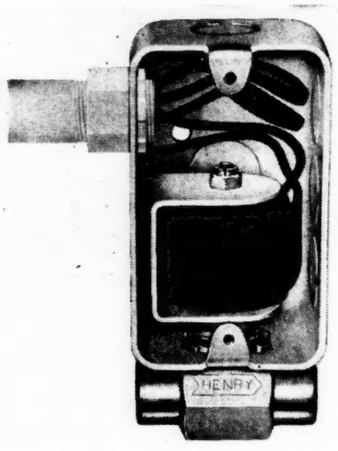
NAME — Oiltight and dustproof plugging switch.
PURPOSE—Designed to remove plugging power from a motor at the correct moment.
FEATURES—Contacts are of silver and they can handle directly the coil current of a 150-ampere contactor. Clean contact surfaces are maintained by the wiping action which results from the use of spring bronze contact strip. Adjustment of contact operating speed can be made by changing the tension of two springs with a screwdriver. Switches have grease packed ball bearings for operating the shaft and magnet assembly. Gaskets are used under the cover of the switch



and around the shaft to prevent dust from entering. The maximum continuous rotating speed of the unit is 1,800 rpm. Two sizes of spring provide for operating speed over ranges of 40 to 140 rpm, and 140 to 750 rpm. Unit can be had in flange mounted form for built in applications, or surface mounted for general use.
MADE BY—*General Electric Co., Schenectady, N. Y.* **236**

Solenoid Valve

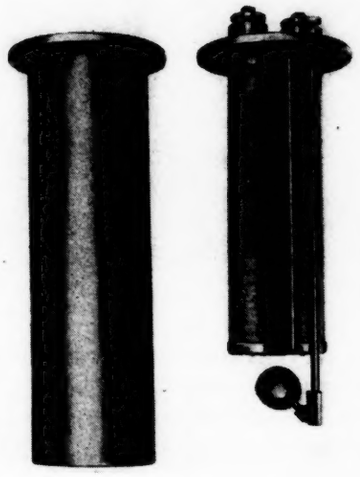
NAME—Henry solenoid valve, type SV-11.
PURPOSE—Designed to meet the need of a solenoid valve for small capacity installations.



FEATURES—Valve is incorporated in a standard outlet box with room provided for making connections with wiring from thermostat or other electrical devices. On special order valves can be mounted in larger outlet boxes. Small screw holes are provided in the back of the box to facilitate mounting. Valve will open against rated opening pressure at 90% of the rated voltage. Coils are double dipped and baked in insulating varnish.
MADE BY—*Henry Valve Co., 3260 West Grand Avenue, Chicago 51, Ill.* **237**

Oil Separator

NAME—Acme oil separator.
PURPOSE — Separation of oil from gas.



FEATURES — Separator contains double oil separating cartridges; a multiple layer of mesh screen on both the gas inlet and the gas outlet. It is also easily accessible for servicing.
MADE BY — *Acme Industries, Inc., Jackson, Mich.* **238**

Electrostatic Precipitator

NAME — Koppers-Elex electrostatic precipitator.
PURPOSE — Removal of solid or liquid particles from plant discharge gases.
FEATURES—Equipment is designed to remove boiler fly ash, tar, acid mists, soda ash, fluorides, dust, and other substances. Precipitators are engineered to the needs of individual plants to provide any desired efficiency.
MADE BY—*Koppers Company, Inc., Pittsburgh 19, Pa.* **239**

STATEMENT OF THE OWNERSHIP, MANAGEMENT, CIRCULATION, ETC., REQUIRED BY THE ACTS OF CONGRESS OF AUGUST 24, 1912, AND MARCH 3, 1933, OF HEATING AND VENTILATING, published monthly at New York 13, N. Y., for October 1, 1946.

State of New York,
County of New York, ss.:

Before me, a Notary Public in and for the State and County aforesaid, personally appeared Edgar A. Becker, who, having been duly sworn according to law, deposes and says that he is the Treasurer of The Industrial Press, publishers of HEATING AND VENTILATING, and that the following is, to the best of his knowledge and belief, a true statement of the ownership, management, etc., of the aforesaid publication for the date shown in the above caption, required by the Act of August 24, 1912, as amended by the Act of March 3, 1933, embodied in section 537, Postal Laws and Regulations, to wit:

1. That the names and addresses of the publisher, editor, managing editor, and business managers are:

Publisher, The Industrial Press, 148 Lafayette St., New York 13, N. Y.

Editor, Clifford Strock, 148 Lafayette St., New York 13, N. Y.

Managing Editor, none.

Business Managers, Robert B. Luchars, 148 Lafayette St., New York 13, N. Y.; Edgar A. Becker, 148 Lafayette St., New York 13, N. Y.; Harold L. Gray, 148 Lafayette St., New York 13, N. Y.

2. That the owner is:

The Industrial Press, 148 Lafayette St., New York 13, N. Y.; Robert B. Luchars, 148 Lafayette St., New York 13, N. Y.; Edgar A. Becker, 148 Lafayette St., New York 13, N. Y.; Franklin D. Jones, 148 Lafayette St., New York 13, N. Y.; Walter E. Robinson, 148 Lafayette St., New York 13, N. Y.; Charles O. Herb, 148 Lafayette St., New York 13, N. Y.; Harold L. Gray, 148 Lafayette St., New York 13, N. Y.; Helena E. Oberg, 65-82nd St., Brooklyn 9, N. Y.; Wilbert A. Mitchell, 28 Harlow Road, Springfield, Vt.; First National Bank and Trust Co. of Montclair and Robert B. Luchars, Trustees (Beneficiaries unknown), Upper Montclair, N. J.; First National Bank and Trust Co. of Montclair and Leigh Roy Urban, Trustees (Beneficiaries unknown), Upper Montclair, N. J.; First National Bank and Trust Co. of Montclair and Kenneth D. Ketchum, Trustees (Beneficiaries unknown), Upper Montclair, N. J.; Paterson Savings Institution, Trustee (Beneficiaries unknown), Paterson, N. J.; Robert Luchars Urban, 51 Ellery St., Cambridge 38, Mass.

3. That the known bondholders, mortgagees, and other security holders owning or holding 1 per cent or more of total amount of bonds, mortgages, or other securities are:

Charlotte B. Baldwin, 420 Clinton Ave., Brooklyn, N. Y.; John Connolly, 148 Lafayette St., New York 13, N. Y.; Franklin D. Jones, 148 Lafayette St., New York 13, N. Y.; Robert B. Luchars, 148 Lafayette St., New York 13, N. Y.; Louis Pelletier, 148 Lafayette St., New York 13, N. Y.; Elizabeth Y. Urban, 163 Western Drive, Longmeadow 6, Mass.; Helen L. Ketchum, 231 King St., Cohasset, Mass.; Wilbert A. Mitchell, 28 Harlow Road, Springfield, Vt.; Henry V. Oberg, 1317 Hillcrest Road, Lancaster, Pa.

4. That the two paragraphs next above, giving the names of the owners, stockholders, and security holders, if any, contain not only the list of stockholders and security holders as they appear upon the books of the company but also, in cases where the stockholder or security holder appears upon the books of the company as trustee or in any other fiduciary relation, the name of the person or corporation for whom such trustee is acting, is given; also that the said two paragraphs contain statements embracing affiant's full knowledge and belief as to the circumstances and conditions under which stockholders and security holders who do not appear upon the books of the company as trustees, hold stock and securities in a capacity other than that of a bonafide owner; and this affiant has no reason to believe that any other person, association, or corporation has any interest direct or indirect in the said stock, bonds, or other securities than as so stated by him.

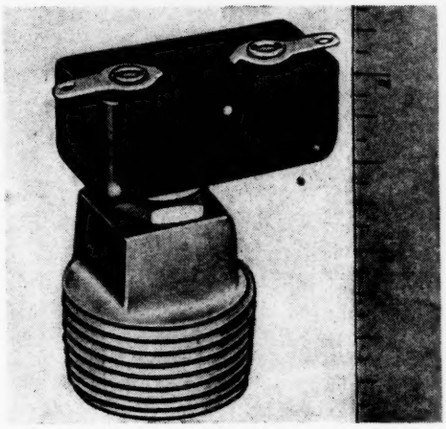
EDGAR A. BECKER, Treasurer

Sworn to and subscribed before me this 1st day of October, 1946.

CHARLES P. ABEL,
Notary Public, Kings County No. 321; Kings Register's No. 116-A-7; N. Y. County No. 325; N. Y. Register's No. 153-A-7. My commission expires March 30, 1947.

Switch

NAME—Cook Pipe Plug switch.
PURPOSE — For detecting the pressure changes in any pipe system and converting them to electrical impulses. Uses include control of pressure cookers, heating elements, water circulating pumps, water supply systems, and so forth.



FEATURES — Has actuating bellows of tinned phosphor-bronze. Diaphragms are completely enclosed.
SIZES AND CAPACITIES—Operates at pressures from 10 to 50 lb per sq in. with a fixed differential of between 8 and 10 lb per sq in. Current capacity: 10 amps at 120 volts a-c. Pipe plug mounting: ¾-in. pipe tap standard, larger size optional.
MADE BY — *Diaphlex Division of Cook Electric Company, Chicago 14, Ill.* **240**

Insulation

NAME—Cellular rubber insulation.
PURPOSE—For use in insulating refrigerators and freezing units.
FEATURES—Cellular rubber can be made in a hard or soft form and is said to resist fire, rot, acids, oil, vermin and termites. Makers contend it is especially suitable for refrigerator trucks. During the war the material was used to protect radar equipment and in the wings of airplanes as supports for fuel tanks.
MADE BY — *United States Rubber Co., Rockefeller Center, New York, N. Y.* **241**

Blind Rivet

NAME—Cherry blind rivet.
PURPOSE—Monel metal fastener.
FEATURES—Identical in construction with the other members of the Cherry blind rivet family, it is made in self-plugging and pull-through hollow types and two types of heads. Aside from the monel rivet announced, it is also available in aluminum, brass and steel.
SIZES AND CAPACITIES—⅛, 5/32, 3/16, ¼, and 9/32 in. diameters and in a wide variety of grip lengths.
MADE BY—*Cherry Rivet Company, 231 Winston Street, Los Angeles 13, Calif.* **242**

December, 1946

TO OBTAIN FURTHER INFORMATION ON NEW EQUIPMENT

On the list below, circle the item number of the equipment in which you are interested, using the number found at the end of each item.

222	223	224	225	226	227	228	229
230	231	232	233	234	235	236	237
238	239	240	241	242			

Print your name and address, detach and mail to
Editor, HEATING AND VENTILATING, 148 Lafayette St., New York 13, N. Y.

Name.....

Title (Must be shown).....

Firm (Must be shown).....

(For prompt service, title and firm name must be shown above.)

Business Address.....

City.....**Zone**.....**State**.....12/46

Degree-Days for October, 1946

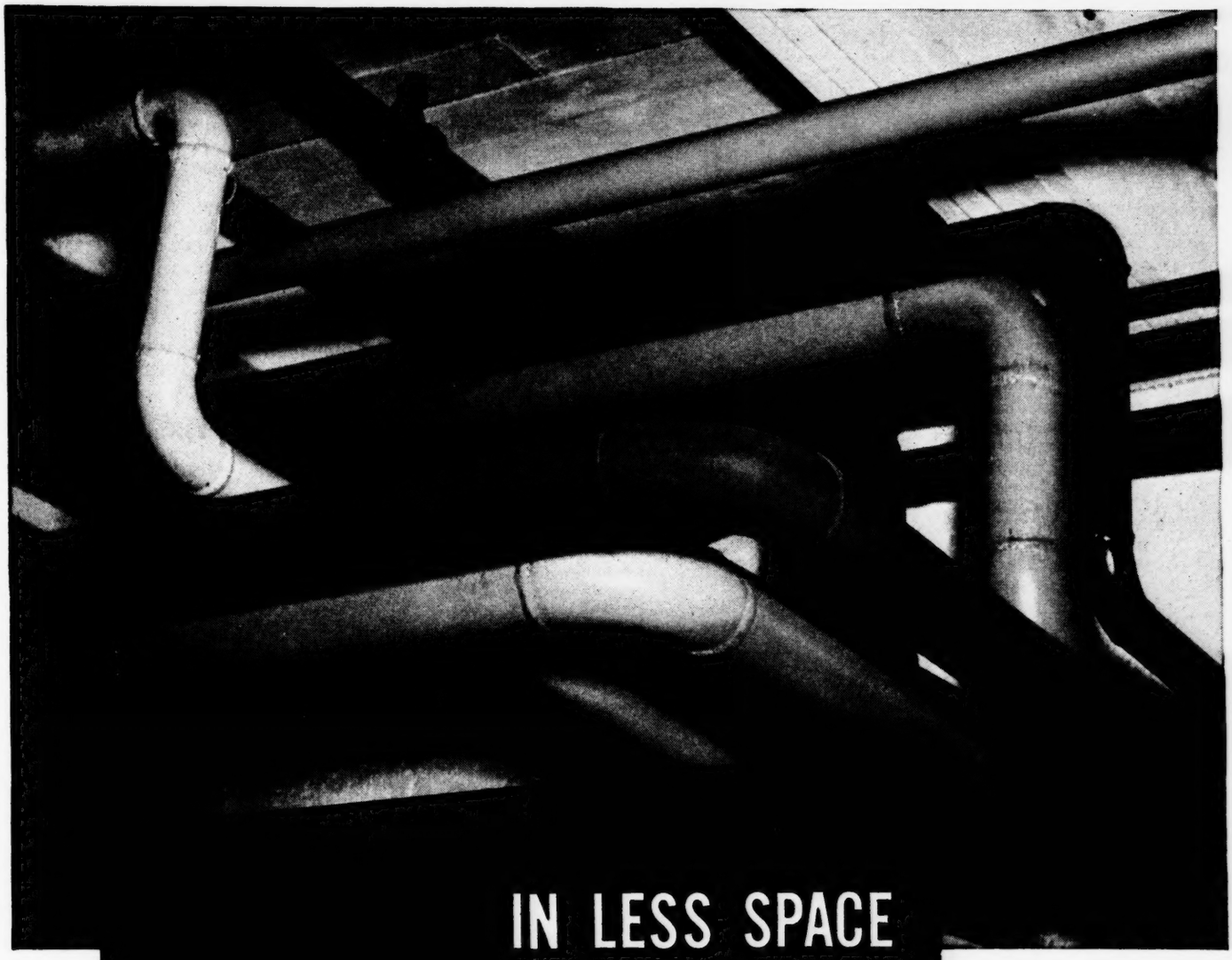
HEATING AND VENTILATING's 20th Year of Publication of Monthly Degree-Day Data

City	October			Cumulative, September 1 to October 31			Heating Season[1]
	1946	1945	Normal	1945-46	1944-45	Normal	Normal
Abilene, Texas	44	90	0	49	127	0	2061
Albany, New York	345	502	446	436	613	518	6580
Albuquerque, New Mexico	273	205	273	285	258	273	4298
Alpena, Michigan	441	579	570	657	579	792	8299*
Anaconda, Montana	897	495	647	1277	893	982	8357**
Asheville, North Carolina	214	300	276	249	318	276	4232
Atlanta, Georgia	83	132	96	97	133	96	2890
Atlantic City, New Jersey	138	229	254	157	248	254	5176
Augusta, Georgia	62	81	16	62	81	16	2161
Baker, Oregon	739	449	552	1034	777	810	7163
Baltimore, Maryland	134	206	223	145	225	223	4533
Billings, Montana	695	407	363	906	706	436	7119
Binghamton, New York	295	468	453	397	582	561	6808
Birmingham, Alabama	95	140	0	105	141	0	2352
Bismarck, North Dakota	748	574	626	1011	904	848	9192
Block Island, Rhode Island	212	325	279	260	371	279	5788
Boise, Idaho	615	307	434	782	534	536	5552
Boston, Massachusetts	234	368	363	291	439	411	6045
Bozeman, Montana	805	499	651	1114	885	987	8521**
Buffalo, New York	272	470	419	381	595	494	6822
Burlington, Vermont	393	565	481	522	736	625	7514
Butte, Montana	932	578	655	1365	1024	999	8272
Cairo, Illinois	99	190	158	109	206	158	3909
Canton, New York	404	566	557	571	776	746	8020
Charles City, Iowa	337	481	476	507	702	569	7588
Charleston, South Carolina	20	45	30	20	45	48	1769
Charlotte, North Carolina	111	139	115	117	142	139	3120
Chattanooga, Tennessee	153	197	105	161	198	105	3118
Cheyenne, Wyoming	681	489	605	938	848	845	7466
Chicago, Illinois	176	362	326	232	465	402	6077
Cincinnati, Ohio	155	292	254	169	316	254	4684
Cleveland, Ohio	223	389	366	285	464	393	6155
Columbia, Missouri	152	280	260	182	350	260	4922
Columbia, South Carolina	64	88	31	65	88	31	2364
Columbus, Ohio	187	347	313	222	378	313	5398
Concord, New Hampshire	436	556	484	578	720	652	7353
Concordia, Kansas	263	264	276	338	375	276	5315
Dallas, Texas	47	75	0	47	93	0	2256
Davenport, Iowa	201	388	363	274	511	363	6289
Dayton, Ohio	(a)	408	273	(a)	455	273	5264
Denver, Colorado	488	315	428	587	534	500	5874
Des Moines, Iowa	261	350	357	364	474	357	6384
Detroit, Michigan	228	433	400	313	542	442	6490
Devils Lake, North Dakota	767	662	706	1082	1052	982	9970
Dodge City, Kansas	299	254	276	363	372	276	5035
Dubuque, Iowa	265	435	409	379	596	445	6790
Duluth, Minnesota	611	654	626	915	998	896	9483
Eastport, Maine	411	559	543	610	798	819	8520**
Elkins, West Virginia	352	467	378	476	529	441	5697
El Paso, Texas	38	84	40	42	112	40	2428
Ely, Nevada	785	531	(a)	1053	842	(a)	(a)
Erie, Pennsylvania	211	381	378	284	464	414	6273
Escanaba, Michigan	477	625	591	749	888	834	8771*
Evansville, Indiana	185	277	155	219	302	155	4244
Fort Smith, Arkansas	103	152	62	109	160	62	3147
Fort Wayne, Indiana	262	464	347	349	566	347	5925
Fort Worth, Texas	42	66	0	42	88	0	2148
Fresno, California	140	38	19	140	42	19	2334
Galveston, Texas	6	24	0	6	24	0	1016
Grand Junction, Colorado	477	279	369	539	370	369	5548
Grand Rapids, Michigan	242	462	419	336	612	476	6535
Green Bay, Wisconsin	375	562	505	560	784	637	7825
Greensboro, North Carolina	166	229	143	186	240	143	3529
Greenville, South Carolina	110	148	146	120	152	146	3380
Harrisburg, Pennsylvania	222	345	329	269	402	329	5375
Hartford, Connecticut	278	432	344	330	509	392	6036
Hatteras, North Carolina	22	55	0	22	55	0	2571
Havre, Montana	741	498	620	1020	851	890	8700
Helena, Montana	814	486	614	1127	865	865	7898
Houston, Texas	13	33	0	13	33	0	1157
Huron, South Dakota	581	462	526	774	702	649	8004
Indianapolis, Indiana	153	323	298	192	370	298	5298
Jackson, Miss.	44	97	(a)	44	97	(a)	(a)
Kansas City, Missouri	159	253	219	196	315	219	4956
Kewanee, Illinois	215	282	334	263	351	412	6139
Knoxville, Tennessee	155	217	183	167	217	183	3670
La Crosse, Wisconsin	370	537	462	570	775	558	7322
Lander, Wyoming	789	482	654	1055	847	930	7947

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with nine exceptions, based on local weather bureau reports. Exceptions are Utica, Lewiston, Kewanee, Sioux City, and Jackson, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y.; Norman E. Ross, Bursar, Bates College, Lewiston, Me.; J. M. Hartman, Engineering Department, Kewanee Boiler Corp., Kewanee, Ill.; Alfred R. Wagstaff, Engineer, Sioux City Foundry and Boiler Co., Sioux City, Iowa, and H. M. Ludlow, Box 1368, Jackson, Miss., respectively; Anaconda, Bozeman, Butte and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table concluded on page 104]



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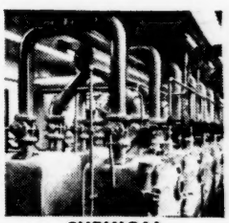
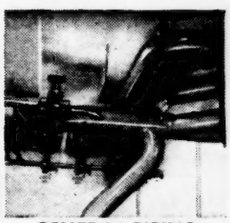
PIPING PERMANENCE SINCE 1927

PETROLEUM

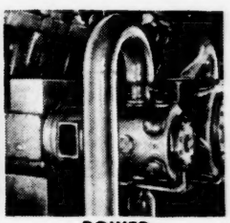


CHEMICAL

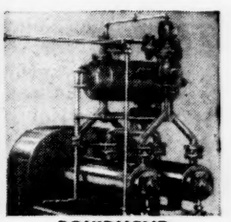


GENERAL PIPING

SHIPBUILDING



POWER



EQUIPMENT

Degree-Days for October, 1946 (Concluded)

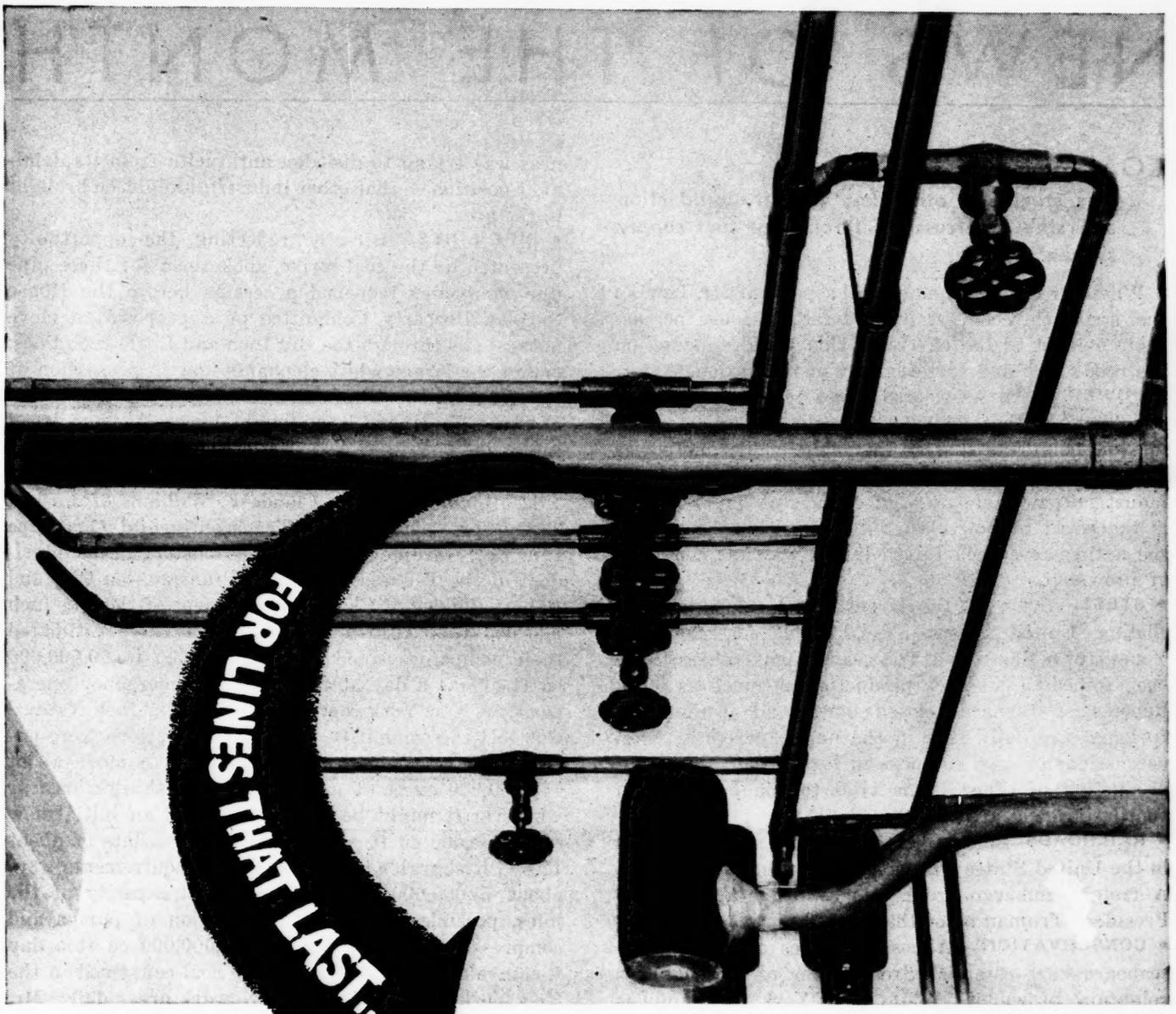
HEATING AND VENTILATING'S 20th Year of Publication of Monthly Degree-Day Data

City	October			Cumulative, September 1 to October 31			Heating Season[1]
	1946	1945	Normal	1945-46	1944-45	Normal	Normal
Lansing, Michigan	293	531	481	421	711	595	7048
Lewiston, Maine	459	599	501	619	794	666	7707
Lincoln, Nebraska	280	287	316	347	399	316	5999
Little Rock, Arkansas	109	144	47	114	157	47	2811
Livingston, Montana	786	441	523	1088	802	750	7205
Los Angeles, California	75	29	0	75	29	0	1504
Louisville, Kentucky	140	238	186	159	250	186	4180
Lynchburg, Virginia	196	251	220	228	266	220	3980
Macon, Georgia	95	98	6	102	98	6	2201
Madison, Wisconsin	306	496	459	438	683	546	7429
Marquette, Michigan	439	608	567	700	889	792	8693*
Memphis, Tennessee	106	161	62	111	172	62	2950
Meridian, Mississippi	64	94	22	66	94	22	2160
Milwaukee, Wisconsin	321	440	450	461	610	534	7245
Minneapolis, Minnesota	433	506	481	628	733	574	7850
Modena, Utah	647	410	517	(a)	588	673	6562
Montgomery, Alabama	36	67	0	38	67	0	1884
Nantucket, Massachusetts	275	325	335	350	381	398	5957
Nashville, Tennessee	128	217	136	135	224	136	3507
New Haven, Connecticut	264	387	360	310	451	399	5895
New Orleans, Louisiana	4	12	0	4	12	0	1024
New York, New York	142	276	272	161	309	322	5274***
†Nome, Alaska	702	660	744	702	660	744	14580**
Norfolk, Virginia	70	107	99	71	107	99	3350
North Head, Washington	445	405	366	679	693	621	5452**
North Platte, Nebraska	485	316	434	588	513	476	6366
Oakland, California	211	148	183	285	223	273	3143**
Oklahoma City, Oklahoma	85	130	105	101	175	105	3613
Omaha, Nebraska	273	315	329	357	431	329	6131
Oswego, New York	286	455	459	398	589	573	7088
Parkersburg, West Virginia	181	305	285	202	339	285	4775
Peoria, Illinois	216	418	381	292	520	384	6109
Philadelphia, Pennsylvania	125	245	235	137	276	271	4737***
Phoenix, Arizona	24	2	0	24	2	0	1405
Pittsburgh, Pennsylvania	193	340	313	225	402	313	5235
Pocatello, Idaho	654	397	499	869	704	655	6655
Portland, Maine	436	593	477	582	771	639	7218
Portland, Oregon	389	288	332	505	399	437	4469
Providence, Rhode Island	226	352	348	276	412	411	6015
Pueblo, Colorado	420	315	388	503	491	394	5514
Raleigh, North Carolina	115	154	130	120	154	130	3234
Rapid City, South Dakota	696	416	515	908	701	659	7118
Reading, Pennsylvania	185	312	344	214	356	344	5389
Red Bluff, California	160	54	(a)	161	59	(a)	(a)
Reno, Nevada	604	353	453	774	548	597	5892
Richmond, Virginia	131	193	161	142	196	161	3695
Rochester, New York	268	467	431	378	605	503	6732
Roseburg, Oregon	457	287	350	575	393	461	4428
Roswell, New Mexico	111	179	186	132	237	186	3484
Sacramento, California	146	42	71	148	55	71	2653
St. Joseph, Missouri	188	287	248	232	369	248	5161
St. Louis, Missouri	117	223	205	141	270	205	4585
Salt Lake City, Utah	579	307	388	716	518	406	5555
San Antonio, Texas	23	54	0	23	59	0	1202
San Diego, California	55	12	34	55	13	34	1645
Sandusky, Ohio	204	368	353	248	429	353	6208
San Francisco, California	169	121	146	286	233	260	3264**
Sault Ste. Marie, Michigan	502	669	623	788	1023	899	9285**
Savannah, Georgia	25	46	0	25	46	0	1490
Scranton, Pennsylvania	266	434	397	334	517	457	6129
Seattle, Washington	436	313	394	577	484	580	4934**
Sheridan, Wyoming	700	432	636	922	744	900	8008
Shreveport, Louisiana	52	81	0	53	81	0	1938
Sioux City, Iowa	401	392	415	529	584	448	6898
Spokane, Washington	643	440	515	851	706	707	6355
Springfield, Illinois	146	282	282	186	348	282	5373
Springfield, Missouri	156	262	217	197	317	217	4428
Syracuse, New York	324	467	425	447	601	521	6893
Tacoma, Washington	490	377	400	682	614	607	5181**
Terre Haute, Indiana	152	307	233	191	351	233	4872
Toledo, Ohio	249	452	369	328	544	378	6077
Topeka, Kansas	184	246	236	233	317	236	4969
Trenton, New Jersey	170	306	242	194	350	242	4933
Utica, New York	329	548	430	444	679	612	6796
Valentine, Nebraska	584	386	490	740	637	583	7039
Walla Walla, Washington	440	252	332	543	388	362	4808
Washington, D. C.	138	240	251	149	256	251	4626
Wichita, Kansas	167	227	192	203	292	192	4673
Williston, North Dakota	756	573	660	1048	977	930	9323
Winnemucca, Nevada	659	380	507	832	623	699	6427**
Yakima, Washington	590	315	437	758	451	557	5599

(a) Data not available.
†Nome data are for September.

*Includes August.
**Includes July and August.
***New 48-year normal covering 1898 to 1946

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.



REVERE COPPER WATER TUBE

WHEN you decide on Revere Copper Water Tube for water or heating lines in any building, you are safeguarding the installation against many kinds of trouble for a long time to come. Naturally this tube is rust-proof. And its smooth gun-barrel interior finish insures an unrestricted flow of water, both in water lines and in high velocity forced circulation hot water heating systems. Joints made with either soldered or compression fittings help further to cut down friction loss.

Made for heating, water supply, air conditioning and other services in all types of buildings, Revere Copper Water Tube is stamped with the Revere name and the type at regular intervals. Look for these identification marks—they insure full wall thickness and the close gauge tolerances so essential for tight sweated joints.

You can specify or install such long-lived Revere materials as Copper Water Tube and Red-Brass Pipe; Sheet Copper for tanks, ducts, pans and trays; Dryseal Copper Refrigeration Tube (dehydrated and sealed); Copper oil burner, heat control and capillary tubes.

Revere materials are handled by Revere Distributors in all parts of the country. The Revere Technical Advisory Service, Architectural, is always ready to serve you. Call your Revere Distributor.

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NEWS OF THE MONTH

COAL STRIKE

cuts steel and other key industry production and stirs repercussions throughout fuel supply system.

Whatever the legal and ethical aspects of Mr. Lewis's coal strike may be, certain hard facts become increasingly evident in the effects of this work stoppage on the production and fuel economy of the nation.

• **POWER.** — An Associated Press report from Pittsburgh states that 25,000,000 workers may be made idle and points out that, in the public utility and power fields, coal supplies 95% of locomotive power, 55% of industrial power and 62% of electric power. So acute is the effect of power curtailment, cities throughout the nation have been forced into a wartime condition of brownout.

• **STEEL.**—Some of the largest steel companies, including United States Steel, Bethlehem, Republic, Youngstown Sheet and Tube, and many others, have been forced to cut back production as much as 50%. Reports of furnaces closed down and workers on furlough are daily fare in the newspaper press. Repercussions of steel cutbacks in forcing curtailment of production of other items vital to the heating and ventilating field are obvious.

• **RAILROADS.**—Passenger service on most railroads in the United States was cut as early as November 25. A freight embargo order was awaiting approval by President Truman as of that date.

• **CONSERVATION.**—Almost all cities on the eastern seaboard have established conditions of more or less voluntary brownout. While New York fuel supplies are fairly large, conservation steps were taken immediately in the appointment of a six-man fuel committee, and fuel conservators have been appointed by most of the eastern states.

• **STOKERS.**—The Stoker Manufacturers Association, at a meeting of its board of directors in Chicago, devoted considerable discussion to the detrimental effect that the coal strike would have on the stoker industry in all parts of the country. The industry has been hampered for some months because of the lack of a supply of bituminous domestic stoker coal, and the present cut-off of production may prove to be a terrific handicap to all segments of the stoker industry. President Walter Sormane expressed his confidence that American public opinion would be brought to bear in forcing peace in the coal fields.

• **ANTHRACITE.**—In Wilkes-Barre normal production was at least temporarily restored to the anthracite industry when the collieries, closed by a sympathy demonstration of mine workers in the northern field, reopened with full force on hand. The General Mine Committee of District 7, while terming the Government's injunction proceedings "Un-American", also decided to check reports that oil burners were being installed in the anthracite field and charged that business was trying to dislodge anthracite from its dominant position so that other industries might be brought to the area.

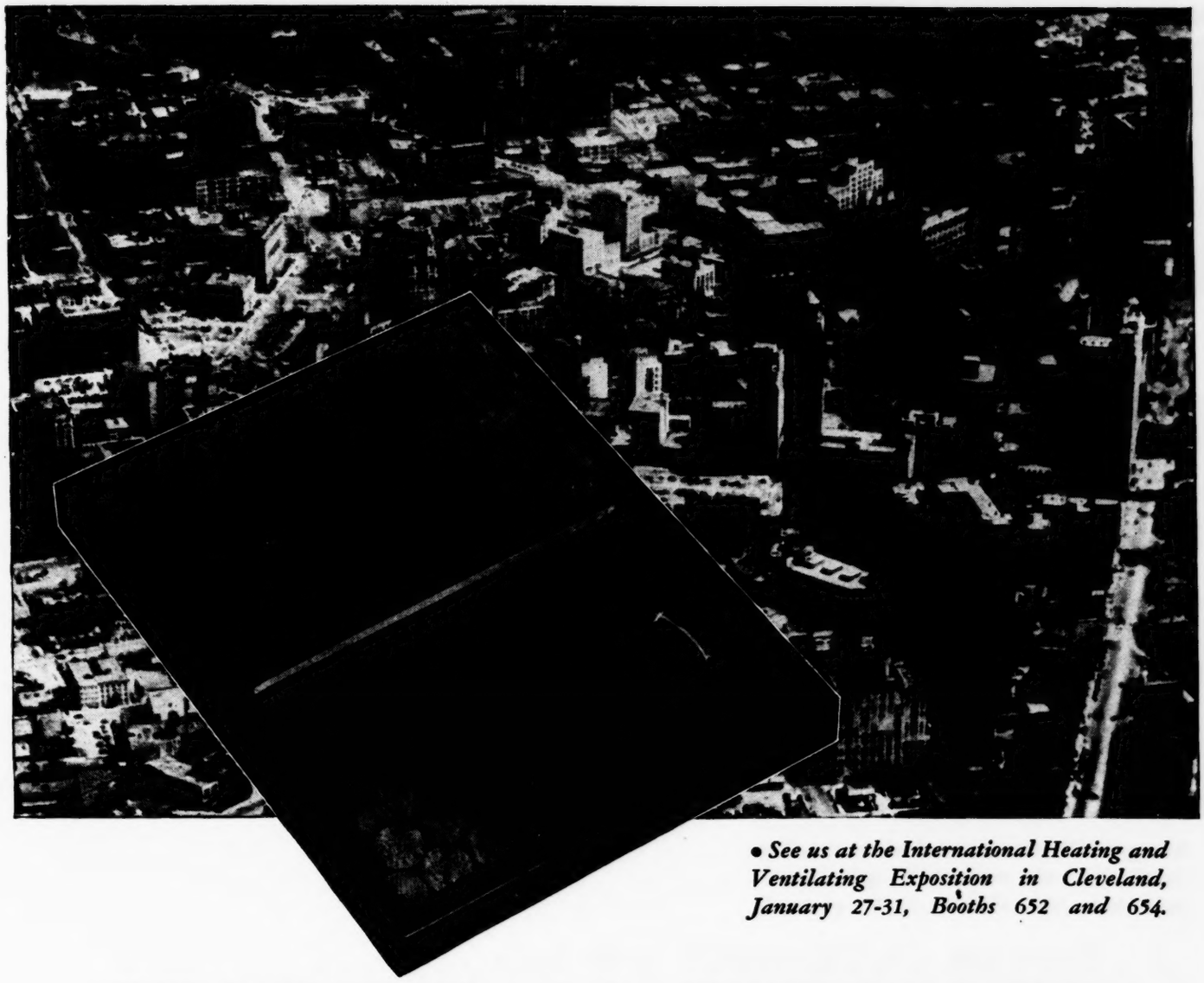
• **PIPE LINES.**—Directly reflecting the opportunity presented by the coal strike, spokesmen for three pipe line companies reopened a session before the House Surplus Property Committee of a proposal to move natural gas through the Big Inch and Little Inch lines, one of the larger white elephants now in possession of the War Assets Administration. Bids for conversion of these lines have been fought by coal and railroad interests.

The three witnesses who appeared before the House Committee were Claude F. Williams of Austin, Texas, president of the Transcontinental Gas Pipe Line Co.; Gardiner Symonds, of Houston, Texas, president of the Tennessee Gas and Transmission Co.; and Robert Bulkley of Cleveland, president of the Big Inch Natural Gas Transmission Co. All three estimated their companies could move 50,000,000 to 90,000,000 cu ft of gas a day at the outset of emergency operations. A New York representative of Big Inch Natural Gas told the committee that after operations were under way the company would be able to move about 140,000,000 cu ft of gas into the Pittsburgh area or wherever it might be needed. He said an initial flow of 90,000,000 cu ft would meet requirements of about 12 of Pittsburgh's industries whose requirements were about 75,000,000 cu ft daily. Total capacity of the lines, possible only with the addition of pumps and compressors, would be about 425,000,000 cu ft a day —equivalent to about 3% of the coal consumed in the New York, Philadelphia, and Newark areas daily. Mr. Williams told the committee that if the company's offer of $92 million cash for the Big and Little Inch lines was not accepted by the War Assets Administration, his company would build a line of its own. The lines were built during the war for petroleum transportation at a cost of about 146 million dollars.

Arthur G. Logan, General Counsel for Big Inch Natural Gas, said he had outlined his company's emergency proposal with C. Gerard Davidson, Assistant Secretary of the Interior, who told him the Interior Department had decided to ask that the Texas-East Coast lines be used for natural gas. At press time a 4-month lease had been granted the Tennessee Gas and Transmission Co. to supply Ohio, Indiana, and possibly Pennsylvania.

• **MIDWEST.** — Meanwhile, the Mid-Continent Gas Transmission Co. of St. Paul presented its case before the Federal Power Commission at a 3-cornered battle for natural gas pipe line facilities to serve communities in six middle western states.

Seeking authority to construct 836 miles of 26 inch line from the Hugoton, Kansas, gas field to St. Paul, the recently organized company was the first to be heard by the commission in a composite hearing on the application of three companies to build and extend



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Complete information on DUST-STOP Air Filters will be sent on request. Write for 24-page illustrated booklet—"Air Filtration in Central Systems" (A5.2.1), Owens-Corning Fiberglas Corporation, Department 912, Toledo 1, O. Branches in principal cities.

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lines and expand their facilities. Other companies to be heard are Cities Service Gas Co. of Oklahoma City and the Northern Natural Gas Company of Omaha.

• **STOCK MARKET.**—The coal strike had the effect of a blanket on the New York Stock Exchange as trading receded to 790,000 shares on November 25 against 1,200,000 shares on the Friday preceding.

FORD HEALTH PACT

forestalls strike vote as union ratifies grant of voice on hazards and cancels walkout notice.

Threats of a strike of 65,000 members of Local 600, United Automobile Workers, C.I.O., against the Ford Motor Company were removed when a union meeting voted to ratify an agreement calling for negotiation of plant health problems.

• **FLUORIDE.**—On October 13 the union took a strike vote which led to the filing of the strike notice. In taking the strike vote the union charged that the company was taking advantage of the no-strike clauses of the union contract in forcing workers to take jobs detrimental to their health. The workers objected to the use of sodium fluoride in the manufacture of steel, contending that the fumes are harmful. The company denied that the use of sodium fluoride was hazardous. Tests were made by the State Department of Labor officials but their findings were not conclusive and the company announced its intention to continue the use of sodium fluoride.

• **NEGOTIABLE.**—The strike notice which would have become effective November 29 was withdrawn after an agreement had been drawn up designating the question of health hazards as negotiable. Under agreement the union will be permitted to bring its own health experts into the plant to make examinations of points which may be in process of negotiation. Before the agreement, according to the union, the company took the position that the health question was not covered by contract and was therefore not open to bargaining.

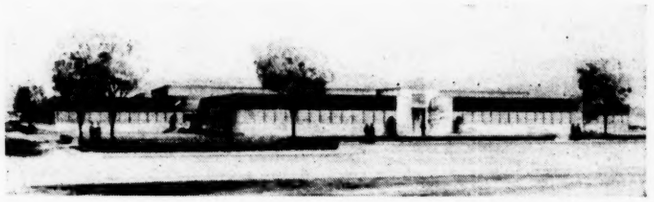
The New York State Department of labor designates all fluorides as hazardous and has established a maximum concentration of two to four mg per cu meter.

NEW FACTORY

to be built by heating specialties manufacturer will be air conditioned throughout and will use various experimental heating systems.

CPA approval has been granted the James P. Marsh Corp., manufacturer of industrial instruments and heating specialties, to start immediate construction of a new factory and office building. The new building will be erected on a five acre tract located at Howard and St. Louis Avenue at Skokie, Illinois, a suburb of Chicago.

• **MODERN.**—The building will cover an area of approximately 100,000 square feet and will be a one story structure of steel and concrete with a monitor type roof. The entire space will be air conditioned and sound-proofed. As manufacturers of heating equipment, the Marsh corporation plans to use the building as a "guinea pig" for extensive research on heating methods. The new plant was designed by the architectural firm of Graham, Andersen, Probst and White and will be erected by the A. L. Jackson Company, general contractors. Location and facilities were planned to provide as nearly ideal working conditions as possible from the standpoint of transportation, material supply, personnel health and comfort, etc.



New factory to test heating systems.

ASRE PAPERS

to be presented at 42nd Annual Meeting, New York City. Program includes transportation symposium.

Five technical sessions have been arranged for the 42nd annual meeting of the American Society of Refrigerating Engineers that will be held at the Pennsylvania Hotel, New York City, December 16-18. One of the highlights is a transportation symposium consisting of seven papers and discussions that will run concurrently with one of the general sessions.

• **PROGRAM.**—Following are some of the papers that will be presented at the convention.

Research Needs in Refrigeration, by Dean Samuel C. Prescott, Massachusetts Institute of Technology, Cambridge, Mass.

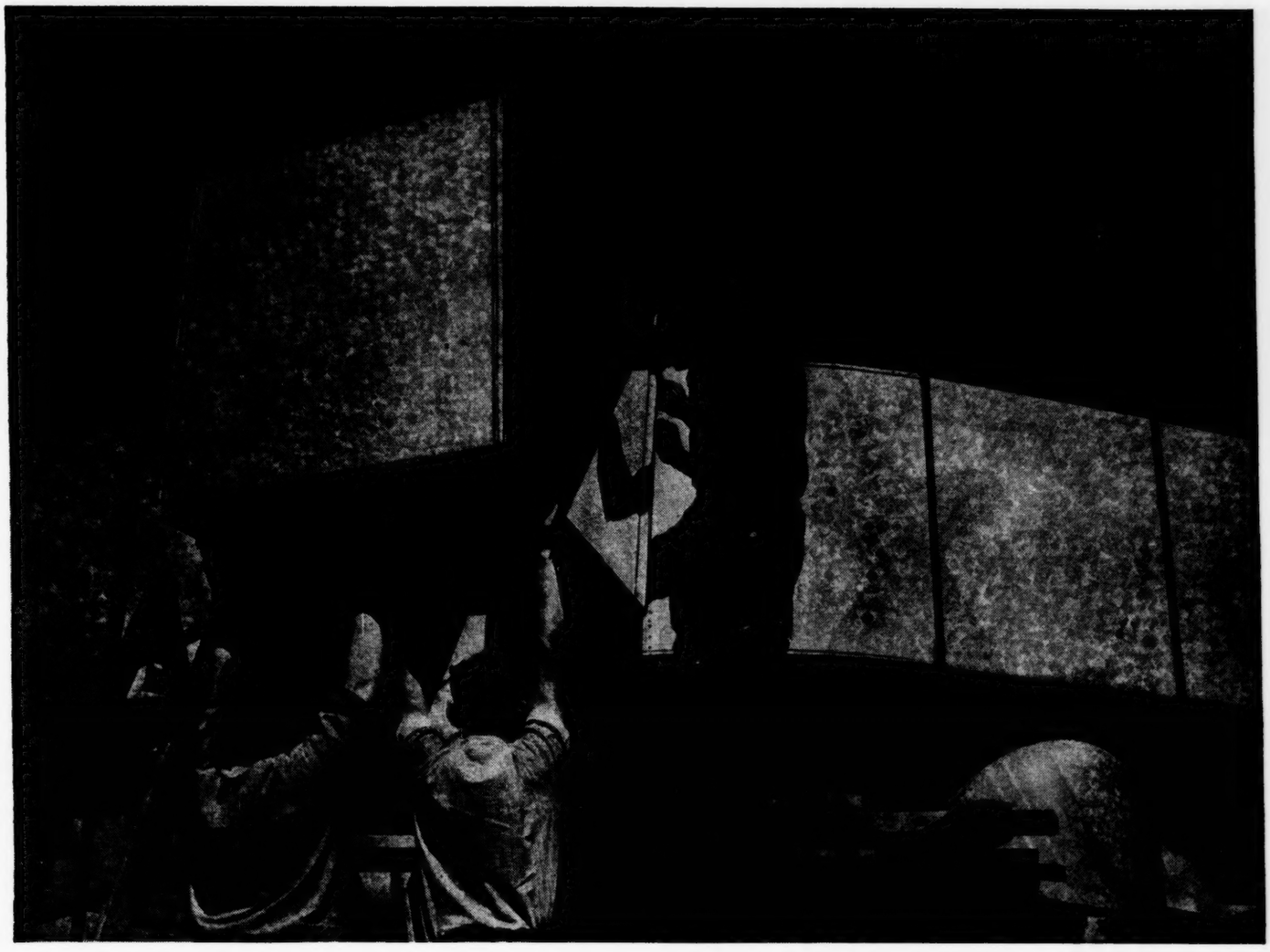
Performance of Small Mechanical Draft Cooling Towers, by Dean T. K. Sherwood, Massachusetts Institute of Technology, Cambridge, Mass., and W. M. Simpson, Servel, Inc., Evansville, Ind.

Low Pressure Liquefaction of Air, by Professor J. H. Rushton, Director, Department of Chemical Engineering, Illinois Institute of Technology, Chicago.

Thermodynamic Properties of the Saturated Liquid and Vapor of Ammonia-Water Mixtures, by George Scatchard, L. L. Epstein, James Waburton and D. V. Cody, Massachusetts Institute of Technology, Cambridge, Mass.

Low Temperature Conductivity Studies, by Frank B. Rowley, R. C. Jordan and Robert M. Lander, University of Minnesota, Minneapolis, Minn.

Condensation of Freon-12 With Finned Tubes; I—Single Horizontal Tubes; II—Multi-tube Condensers, by Donald L. Katz, Ralph E. Hope, Stever C.



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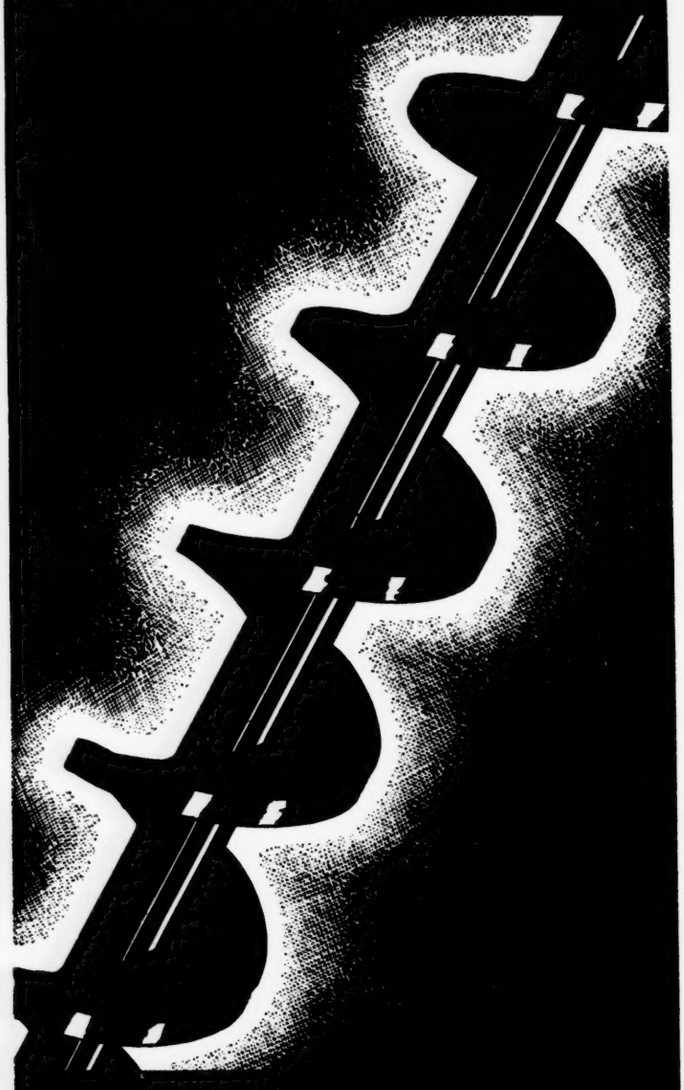

is an extremely versatile material. Not only is it tough and wear resisting, but it is capable of being readily shaped into an infinite variety of sizes and pitches, tapered or constant diameter, right hand or left hand. That's why so many stoker manufacturers use the Crown wrought steel stoker feed screw. They know that Crown can produce a tough, continuously welded wrought steel screw to meet their most exacting specifications.

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Construction of Railway Refrigerator Cars, by R. G. Setzekorn, Mechanical Superintendent, American Refrigerator Transit Company, St. Louis, Mo.

Refrigerated Trailers, by Henry Kirkpatrick, Chief Engineer, Advance Manufacturing Co., Detroit, Mich.

Cargo Refrigeration, by H. E. Parker, Engineer, Central Technical Department, Ship-Building Division, Bethlehem Steel Company, Quincy, Mass.

Refrigeration Requirements Aboard Naval Vessels, by Commander C. V. Hawk, Bureau of Ships, U. S. Navy, Washington, D. C.

RESEARCH PROGRESS

in heating and ventilating outlined by laboratory director at New York meeting while other groups report on various projects.

Research on a broad range of subjects of concern to heating, ventilating, and air conditioning engineers under way at or planned by the Research Laboratory of ASHVE was discussed by Cyril Tasker, Director of the laboratory, at a meeting of the New York Chapter of the Society November 18. Other groups also are reporting investigations of interest to the field.

• **STAFF.**—Mr. Tasker, who is in charge of a staff including three senior engineers, several juniors, clerical and mechanical assistants, told his audience that in its twenty-seven years of existence the Research Laboratory had produced 250 papers. He emphasized that the laboratory belonged to the members of the ASHVE since it is directly financed by the dues from members. Collaborating in this research are twenty technical advisory committees.

• **PROJECTS.**—Research now under way or planned for the near future in the laboratory includes, Director Tasker said, study of air friction in ducts and fittings; performance of barometric dampers, air flow and distribution from slots, jets, and perforated panels, industrial ventilation code analysis; transmission of solar radiation through glass and roofs; standardization of heat transmission coefficients for insulating materials; weather design data analysis; air cleaning devices; and forced convection heat transfer.

• **BLOWER GROUP.**—A new research program will be conducted by the Furnace Blower Manufacturers Association at the University of Illinois. First item on the program will be determination of test methods which will enable the furnace designer to determine the resistance of a combined blower-furnace assembly. Collaborating in the program is the National Warm Air Heating and Air Conditioning Association, whose initial research problems at Urbana, Ill., will be closely linked with those of the furnace manufacturers.

• **SETUP.**—To guide the program in its first stages, an engineering committee has been formed with

High pressure shell and tube condenser for CO_2 refrigerant, replacing mass of double pipe coils shown above.

We design and manufacture heat exchangers for Freon and other refrigerants for refrigeration and air conditioning services. Quick deliveries, Materials available. We recently designed and built eight large Freon condensers in five days. On an urgent West Coast order we designed and built a unit in three days. Two units were built over a weekend for an emergency. Standard units in stock. For quick, economical service, write, wire or phone, Days, Nights, Holidays.

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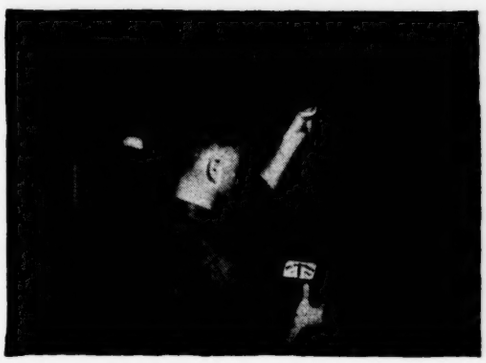

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The Alnor Velometer is built in several standard ranges, 0-200 to 0-6000 fpm, and up to 20 inches static on total pressure. Special ranges available up to 24,000 fpm. Write for velometer bulletin with complete description.

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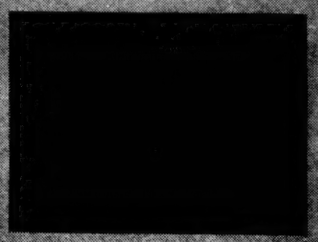

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A proven heating unit, modern in design, incorporating every unit heater advancement. McCord copper spiral fin tube construction creates air turbulence without undue restriction. Improved fan blades deliver large volume of air with minimum air noise. Guaranteed for 150 lbs. saturated steam pressure. Specify McCord.

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A time proven heating unit, modern streamlined in design, quiet, yet its rugged construction adapts it to all types of installations, guaranteed for use on No. 150 p. s. i. steam pressure. All copper heating element, individual spiral fin tubes free to expand without strain on joints. Standard motors with standard bases used. Wide range of capacities.

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Differing radically from conventional valve design, Model 68-B Water Regulating Valve employs a molded composition seal on the valve stem. This seal moves across the polished mirror-smooth stainless steel seat with a "wiping" self-cleaning action that avoids common troubles due to dirt and lime in the water. Laboratory tested in sandy water, this sliding seal and seat showed no sign of wear after the equivalent of many years actual field use — indicating a "life-time" of water regulating service.

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DEPENDABLE *Refrigerant Valves*
FOR AIR CONDITIONING • REFRIGERATION • HEATING

News of the Month

Gordon Kinsman as chairman. Other committee members include Marion Levy and James Hagerman and Gerald Schwartz. The first testing plant will consist of a large plenum chamber containing 4 different flat-plate orifices. The chamber will contain an auxiliary 12-in. blower driven by a ¾-hp motor.

• **DATA.**—Complete measurements of air deliveries, pressure and power input will be included in the data which will be available to manufacturers. A standard test method will be involved and made available to all furnace manufacturers for testing the combination of a blower with a furnace.

• **ANTHRACITE.**—Anthracite Institute has completed arrangements for a four-company research project at Wilkes-Barre for experimentation with various phases of the gasification, oxygenation and pelletizing of the small sizes of anthracite and anthracite fines.

• **EQUIPMENT.**— Each of the four companies will supply facilities which will aid the project. The Wellman Engineering Co. of New York is supplying a small gas producer of a type similar to the several hundred commercial installations that are now used for gasifying the buckwheat and rice sizes of anthracite.

Air Products, Inc. of Emmaus will furnish an individual oxygen generator to supply oxygen instead of air for comparative tests.

J. F. Pritchard & Co., of Kansas City has supplied a commercial sized machine suitable for pelletizing anthracite fines.

The Glen Alden Coal Co. of Wilkes-Barre will furnish space for erection of equipment in view of limited space available in AI's own laboratory. Tests will be conducted on all types of anthracite from the several regions and will be supervised by the Institute.

• **WEST COAST.**—The West Coast Technical Conference on Domestic Gas Research, held in Los Angeles under the joint auspices of the American Gas Association and the Pacific Coast Gas Association, featured a panel discussion on gas water heating research by C. A. Thorpe of the AGA Laboratory staff which was presented in the form of a report prepared by Dr. C. F. Prutton covering research of the galvanic corrosion of dissimilar metals as applied to hot water storage heaters. The tests reported were conducted at the Case School of Applied Science.

• **CENTRAL HEATING.** — Another report at the conference covered the subject of central heating research and was prepared by Robert C. Weast and James A. Morse. This report described heat transfer characteristics of different types of gas flames and also presented highlights of a survey which attempted to determine the popularity, potential demand, and practical effectiveness of gas fired summer-winter hookups designed to provide both heat and hot water during the heating season and hot water during the remainder of the year.

A report on water heater research was presented by W. B. Kirk, assistant chief Research Engineer of the A.G.A. Testing Laboratory. This report described experience with bimetallic thermoelements in automatic gas pilots.

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DIRECT-FIRED Suspended WARM AIR HEATERS

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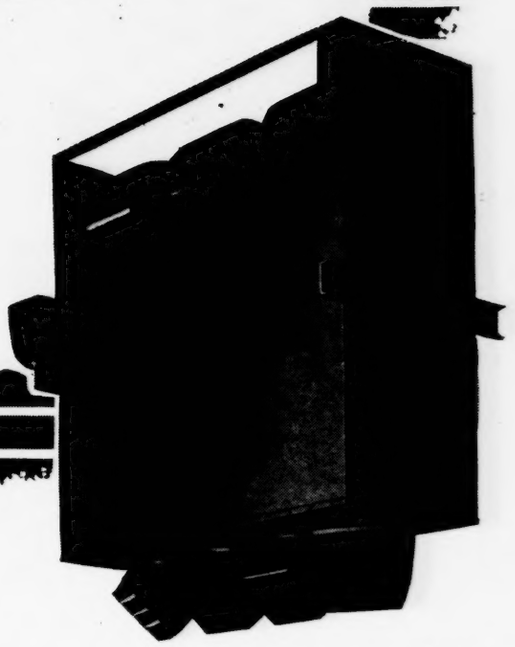
Airtherm suspended models are designed specifically to permit use of all available floor space, while heating your plant efficiently and economically. Control panel can be placed at convenient location near floor. Horizontal or vertical mount. Capacities from 650,000 to 1,950,000 B. T. U.



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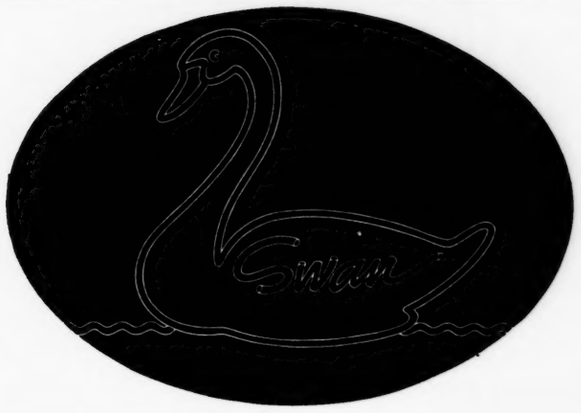

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Many foremost manufacturers, having discovered our specialized skill in our particular line, have entrusted us with a great variety of precision pipe and tube work, including bending, coiling, expanding, flattening, brazing and welding.

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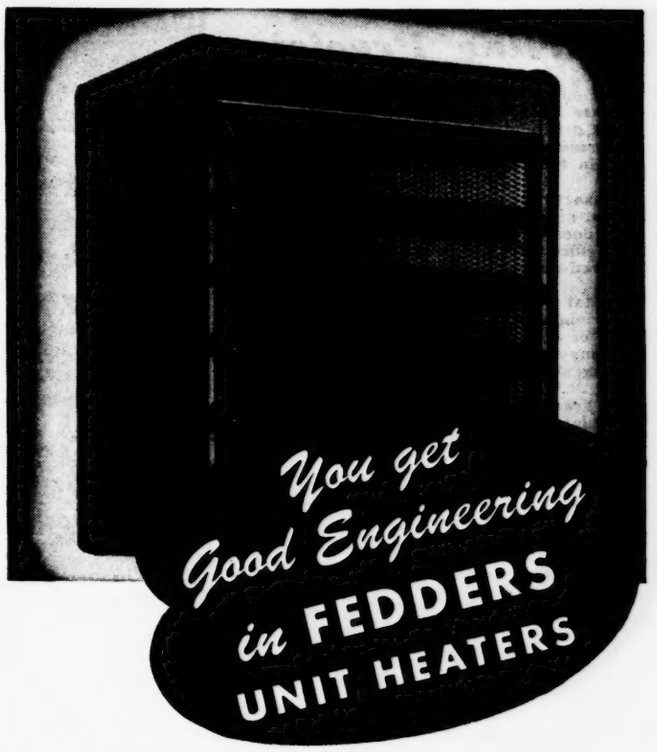

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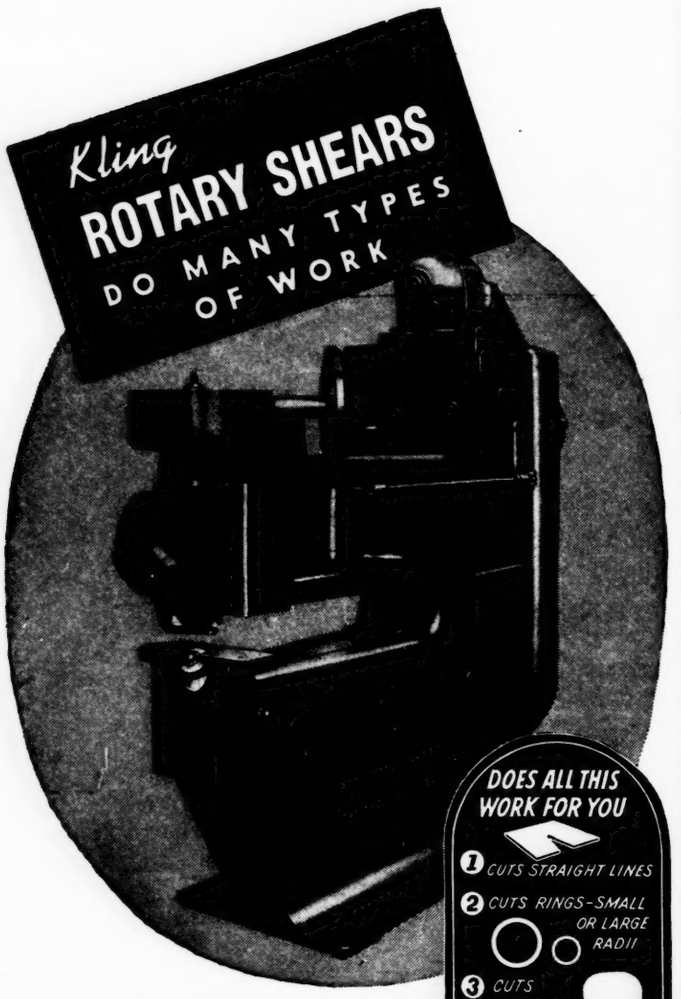

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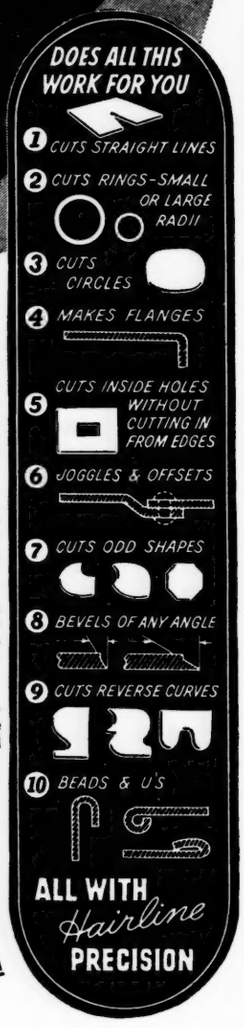
For exacting projects (see illustration at the right), no single unit of metal-working equipment does so many different things so cleanly and efficiently as does the Kling Rotary, pictured above.

In metal working plants, automotive, aviation, home appliance, and other industries, where work of this character is being done, — this machine is held in high regard for its versatility and economy of operation.

This great usefulness is the result of half-a-hundred years of engineering experience which prospective buyers, with reason and respect, applaud.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Send Today for Bulletin No. 245 which will give you specific facts, and specifications. No obligation involved. Users of heavy equipment of Kling Rotary Shears' class will receive a copy of this Bulletin promptly, on request.

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News of the Month

• **PLANT GROWTH.** — Experimental installations of air conditioning apparatus in greenhouses were described at a recent meeting of the Los Angeles chapter of the ASHVE by Art Hess, of English & Lauer, Inc., Los Angeles. The first attempts to air condition greenhouses, according to Mr. Hess, were made in Southern California in the form of experiments by Henry Eversole and the work of L. C. Marshall of the University of California. While these experiments have not progressed to the point of practical application, Mr. Hess believes that air control in greenhouses offers a wide field of developments. He pointed out that dormant plants can be cooled in a refrigerator for 36 hours and replanted immediately. These plants will bloom again the same as though they had been dormant for the winter.

CHEMICAL INDICATOR

extremely sensitive to carbon monoxide is developed by National Bureau of Standards chemists.

A colorimetric indicating gel that will detect and estimate less than one part of carbon monoxide in 500 million parts of air, a sensitivity more than 100 times greater than that attained by former chemical indicators, has been announced by the National Bureau of Standards, gas chemistry sections. The indicator is so sensitive that it is possible to diagnose carbon monoxide poisoning by analyzing exhaled air instead of taking blood samples for an analysis.

• **SENSITIVE.**—The indicating material, yellow in color, is a silica gel impregnated with a complex silico-molybdate compound and catalyzed by means of palladium in the form of a sulphate. The yellow indicator turns various shades of green and blue-green on exposure to very low concentrations of carbon monoxide. The color response is a function of time by concentration of carbon monoxide—a direct parallel to the physiological response to this gas.

• **PORTABLE.**—In use, the indicator gel is sealed in a small glass about five inches long and the approximate diameter of a lead pencil with protecting layers of pure silica gel in the sealed tube. To make a test, the ends of the indicating tube are broken off and the tube inserted in an ordinary 2 oz. rubber aspirator bulb equipped with a special rate controlling valve. The air tube tested is drawn through the tube by squeezing the bulb once and any color that develops is compared with a set of standard color chips. For use in aircraft, tables giving correction factors for altitude will be provided.

• **ORIGIN.**—During 1941 the Royal Aircraft Establishment in Farnborough, England, developed an indicator for use on board aircraft. As this indicator was not entirely satisfactory, the Bureau of Aeronautics, Navy Department, and later the U. S. Army Air Forces requested the National Bureau of Standards to study the R.A.E. gel with the object of improving it or developing a suitable substitute. Starting with this material, the Bureau of Chemists produced an indicator

CAREER OPPORTUNITIES IN SALES OF AUTOMATIC CONTROLS

New developments in construction, heating, air conditioning and industrial control are creating unusual opportunities for men with technical sales ability.

If you've had two years of college plus engineering-sales experience or four years of college with technical training, preferably in heating or air conditioning, investigate this opportunity.

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Another Phillips Product

. . . PILOT OPERATED LIQUID CONTROL VALVE

May be piloted by either float, thermal or electric valve.

The recognized higher efficiency rate of a flooded operated evaporator in a refrigerating or air conditioning system is further emphasized when the evaporator is used in conjunction with Phillips Float Controls. Maximum B. T. U. transfer is assured. These controls respond so instantly to any wide variations in load that their use is indicated on practically all installations. Available capacities: "Freon," 20 to 500 tons; Ammonia, 1 to 2000 tons. For high or low side applications. Write for NEW Bulletin.

The Home of Phillips Regulating Valves and Float Controls Established in 1929

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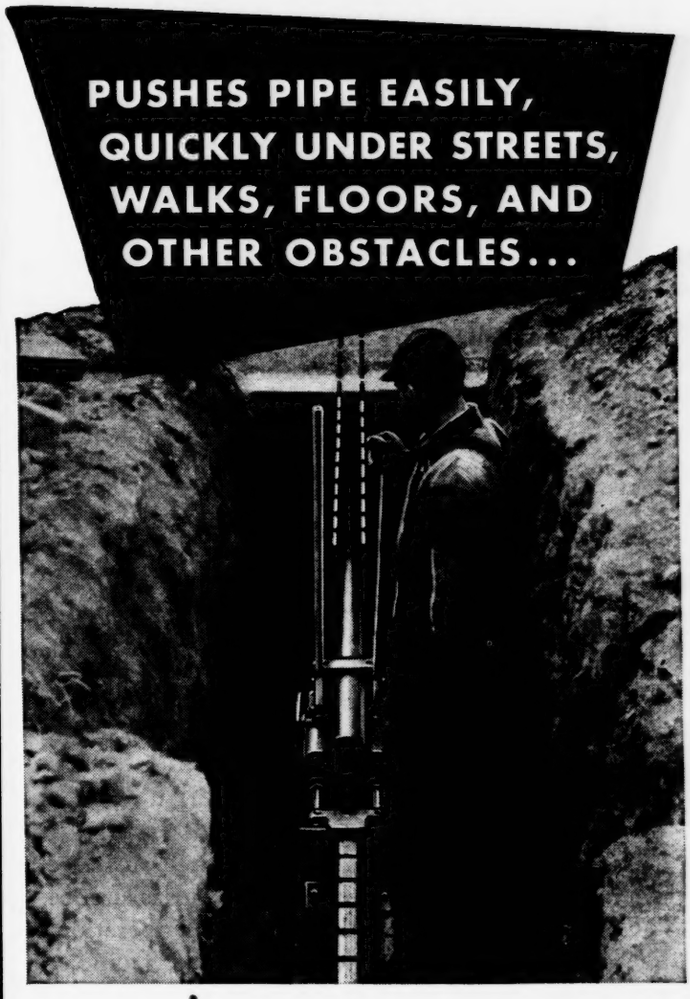


GREENLEE HYDRAULIC PIPE PUSHER

Here's how you can cut costs, save time on underground pipe installations. With a GREENLEE Hydraulic Pipe Pusher *one man* can push pipe through the ground *easily, quickly* — under obstructions!

Save the time, work and inconvenience to all caused by the old methods of tearing up floors, lawns and pavement. Eliminate tedious back-filling, tamping, tunneling, repaving (just a short trench accommodates the pusher).

Save with the GREENLEE which often pays for itself on the first few jobs used. Learn all about this compact, portable, powerful pipe installing equipment . . . get free Folder S-117. Greenlee Tool Co., Division of Greenlee Bros. & Co. 2332 Twelfth St., Rockford, Illinois, U.S.A.

REGISTERED TOOLS

GREENLEE

FOR THE CRAFTSMAN

Let's Talk About COMMUTATORS!

1 Wagner Knows Commutators

Since Wagner introduced the first commercially successful single-phase motor in 1896, the company has manufactured millions of commutators for repulsion-start induction brush lifting motors—far more than any other manufacturer.

2 Wagner Builds Commutators That Last

An almost unlimited number of full-load starts is built into every Wagner commutator. Liberal design provides extra strength, permits refacing several times, if necessary, with safety. Replacement commutators are built to operate at any speed up to and including 3600 RPM without danger of "explosion".

3 Wagner Builds Commutators That Fit

Genuine Wagner replacement commutators are built as an original motor part—your assurance of an accurate all-round fit for correct operation. Segments are accurately milled to facilitate connection of rotor leads.

4 Avail Yourself of Wagner's Experience

Wagner replacement commutators, like all genuine Wagner parts, bring you the benefit of the latest improvements in materials, design and construction.

Commutators Are Available For Immediate Delivery

Wagner motors are world-famous for dependable service. When repairing a Wagner motor, be sure to use *genuine* Wagner motor parts, designed and built to renew that dependability. Get them from one of the 325 authorized service stations displaying this sign.

Every motor repair shop needs this catalog. It helps you determine the catalog number and price of Wagner fast-moving motor parts. Write for Catalog MU-40 today.

M46-25B

Wagner Electric Corporation

EST. WE 1891

6463 Plymouth Ave., St. Louis 14, Mo.

about four times as sensitive, more adaptable to field conditions, and truly colorimetric. The indicators were made only at the National Bureau of Standards and altogether more than half a million were supplied for the military services of the United States, Canada and Great Britain. Patents on the device have been assigned to the Secretary of Commerce.

UNMINED COAL

will be burned in the ground to manufacture gas for industrial use in Alabama experiment.

The Alabama Power Company, with cooperation from the Bureau of Mines, will carry out a full scale experiment in the underground gasification of unmined coal with a view to possible mining economies and the utilization of inaccessible coal veins. The proposal is that producer gas can be generated by burning unmined coal right in the ground.

To carry out the experiment the power company will isolate a coal vein in a new mine at Gorgas, Alabama, set it afire, feed it air and perhaps oxygen, and draw the resulting gases out of the mine, cool and detar them, then store them for use in its power plant nearby.

• **RUSSIANS.**—While this experiment is a new one in the United States, it is known that Russia has had similar projects under way for more than 11 years and claims reasonable commercial success. The gas obtained will be of low heating value compared with normal commercial gas, according to a "Business Week" report.

• **REFINEMENT.**—It is possible to increase the heating value of the gas by adding oxygen to the air, thus reducing the proportion of nitrogen. An obstacle to this procedure in the present experiment is the cost of oxygen. Another possibility is the addition of steam which combines with the carbon at high temperatures to form hydrogen and carbon monoxide. This involves problems in maintaining the fire bed temperature above combustion levels.

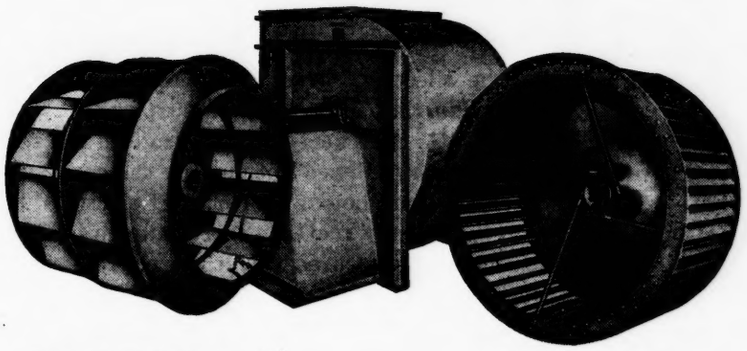
• **LIGNITE.**—Possibilities of the underground gasification method of producing power fuel open a field for the development of all-but-worthless lignite and oil shale deposits.

• **HOW DONE.**—The experiment will be carried out on a coal seam located in a 400-ft neck of land jutting out from a large hill. A trench 20 ft wide will be bulldozed across the base of this neck of land isolating the coal seam, which is 30 in. to 40 in. thick and 40 ft wide. On each side of the vein, horizontal shafts 6 ft in diameter will be driven into the hill from the trench and connected at the down slope end, forming a U-shaped channel.

The plan is to fire the face of the coal vein at the down slope end. Air will be blown in one shaft and the producer gas drawn out the other, as the coal burns toward the cutoff trench. Actual tests will start within 60 to 90 days and may take a year to complete. The project is expected to cost $50 to $75 thousand dollars.

Extra Capacity . .

AIR MOVING EQUIPMENT



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Jan. 27 to 31, 1947

MASSACHUSETTS BLOWER DIVISION

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Just off the Press

This Bulletin, No. 461, gives an up-to-date description of Therm-O-Tile Underground Pipe Conduit and its numerous advantages.

Write for your copy.

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Reg. U. S. Pat. Off.

More for your money. Insulation always dry. Efficiency permanently high. Grades permanently maintained. "Spread footing" foundation. Stronger. First cost is competitive. Lowest final cost. Etc. *Sold and installed by Johns-Manville Construction Units in all principal cities.*

H. W. PORTER & CO., Inc.
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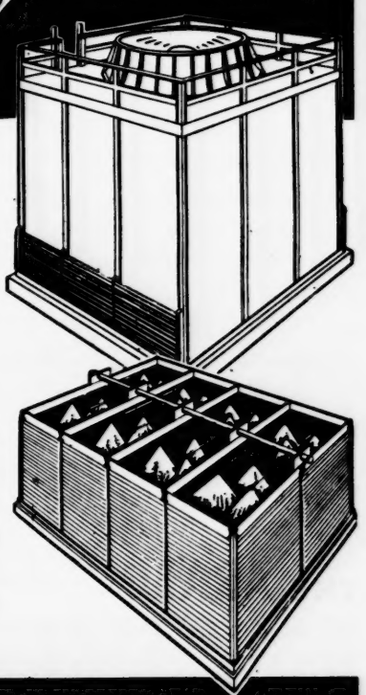
WATER COOLING

Dependability
Economy
Efficiency



Marley Induced Draft and Marley Atmospheric Towers are the product of a quarter century of constant development that has achieved maximum efficiency, economy and dependability.

For any cooling requirement there is a unit in the complete Marley line that accurately fits the need of air-conditioning, refrigeration or processing.

MARLEY COOLING TOWERS

THE MARLEY CO., INC. KANSAS CITY 15, KAN.

Here is the new and easier method of finishing lagging material on pipes, ducts, boilers and hot water tanks. Simply adhere the canvas, asbestos, fiberglas, or other non-conducting material, with Arabol Lagging Adhesive.

This product fills your most exacting requirements for both utility and appearance. It is easy to apply; dries in 4 to 6 hours, leaving a sized finish. And this finish need not be painted at all, unless you desire to add one coat for appearance. Labor costs are lower, because you eliminate the slow process of sewing. Maintenance is no problem—grease, oil, soot and dirt wash off easily. And, finally, it is vermin-proof... fire-retardant, too.

Extensive tests conducted by independent laboratories have shown just what this product can do under exacting conditions. Arabol was found to retain its adhesive strength despite exposure to extreme temperatures, to immersion in water, and to live steam.

Write us today for detailed facts and figures. Don't place open specifications on lagging work—ask for Arabol Lagging Adhesive. You can depend on it to provide a neat installation – satisfactory in every respect. Also—ask about our cork cement for adhering cork to cork on refrigerator lines.

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Adhesives?... ARABOL!

FIVE HEAT PUMPS

installed by Chattanooga utility to be observed in tests of reverse cycle heating and cooling of residences.

Five reverse cycle heat pumps are being installed by the Electric Power Board of Chattanooga in residences in that area for test purposes.

• **ORIGIN.**—The decision to engage in the test installations was occasioned not only by the company's natural interest in promoting all forms of electrical heating but also by a heavy increase in resistance heating loads in the Chattanooga area with their characteristically high demand and low load factor. Low electric service rates and moderate climate influenced the decision to promote the reverse cycle equipment as a forward step in house heating, and as a means of equalizing the summer load by the use of air conditioning facilities.

• **EARTH HEAT.**—The present installations are 3-hp Marvair units, three of which are already operating. Two 5-hp Marvair units are in process of installation. These units operate on a closed circulating water system installed in a deep well.

One of the units now in operation is installed in the home of C. B. Osborne, sales manager of the Electric Power Board. The Osborne house, a six-room brick structure with overhead ceiling insulation and weatherstripped windows and doors, has a heat loss of 66,000 Btu per hr at design conditions. The deep well drilled for the heating unit is 230 ft deep. Water temperature in the Chattanooga area is about 62F.

• **SYSTEM.**—The circulating system involves the use of a U-tube extending down into the well. Water is not pumped from the well, but the water in the system is circulated through the pipe in the well, where a reversible heat transfer takes place.

In the heating cycle, heat from the water in the U-tube is extracted by a refrigeration compressor and coil and is released through a heat exchanger over which air is blown by a circulating fan so that an air temperature of about 105F is obtained. The air is conveyed by ducts like those in the conventional warm air furnace. Before being recirculated the air passes through a filter to remove all dirt and dust. The filter is equipped with germicidal tubes so that not only is it clean but it is also sterilized.

The equipment requires a basement space three ft wide, 7 ft long and 6 ft high. Near the completely enclosed equipment are switches and automatic control mechanisms mounted on a wall panel. Motors used by the system include a water circulating pump powered by a 1/3-hp motor, a refrigeration compressor using a 3-hp motor, and an air circulating system employing a 1/2-hp motor.

• **PLANS.**—The company plans installation of a second type of system which uses a copper coil which is placed in the ground, eliminating the need for a well.

By the first of 1947 the company expects to have in Chattanooga a total of more than 400 electrical resistance heated homes, stores, show rooms and factories.

REFRIGERATED CARGO PLANE

designed by Douglas will be divided to allow two different temperatures in cargo spaces.

The new Douglas DC-4-1037, capable of carrying 22,700 lb of cargo for 1,500 miles, is said to be the largest and safest self-refrigerated cargo carrier now in production.

• **REFRIGERATION.**—Cooling of the new plane is provided by a vapor cycle refrigerating unit using Freon 21 as a refrigerant. The mechanism is operated from the plane's electrical system in flight and from a gasoline powered motor generator on the ground. Thermostatic controls permit the selection of temperature and humidity desired for each of the two compartments. Insulation is provided by a layer of fibreglas identical to the soundproofing of the passenger DC-4 sister ship.

• **HEATING.**—A gasoline combination unit provides heat for either one or both spaces. Dual compartments enable the craft to carry baby chicks at 85F in one part of the fuselage and ripe, fresh fruit at 45F in another part.

• **APPROVED.**—Developed by the mechanical section of the Douglas engineering department of Santa Monica at the suggestion of United Airlines, plans for the refrigerated cargo plane met with the immediate approval of three carriers, according to Douglas.

CLAYTON A. DUNHAM

inventor and manufacturer of heating appliances, dies after long illness.

Clayton A. Dunham, 150 Maple Hill Road, Glencoe, Illinois, chairman of the board of the C. A. Dunham Co., died November 22 at the Evanston hospital, after a long illness. He founded the business in 1903 and served as president until the first of this year, when he retired from that office and became chairman of the board. He was internationally known for his contributions to the art of heating. His life is the story of the development of an idea into a world-wide business.

• **HEATING INVENTOR.**—In 1903 Mr. Dunham, with one assistant, began manufacturing the Dunham fluid-filled disc radiator trap in a small basement shop in Marshalltown, Iowa. It soon became the most widely used trap construction throughout the world and high courts protected his rights as the inventor. A major achievement was the solving of the problem of how to control and distribute the steam supply to a building to meet varying weather conditions. The Dunham differential vacuum heating system announced to the country in 1927 was the culmination of a decade's work of research and was added to the 200 or more patents registered by Mr. Dunham.

Mr. Dunham is survived by his widow, Beth Milnor Dunham; a son, Aubra A.; a daughter, Mrs. Sidney Grove; a brother, Winfield Dunham, and nieces, Miss Gertrude Dunham Miller and Mrs. Charles A. Wright.

Unless you are pretty sharp-eyed, you'll pass by "old man walking stick" as just another dead twig on a tree.

And there are a lot of other things that look different from what they really are . . . for example, certain cut-rate plumbing supplies. The wise plumber will avoid the pitfalls that come with inferior merchandise by relying on the reputable jobber in his vicinity.

Dollar for dollar, the trade-marked, quality merchandise he stocks is your best plumbing buy.

KUHNS BROTHERS CO.
DAYTON 1, OHIO

K
CAST FITTINGS

"K" fittings are carried in stock at Malleable Iron Fittings Co., Branford, Conn.; M. I. F. stocks at Kuhns Bros. Co., Dayton, Ohio.

6 out of 10
manufacturers of original equipment SPECIFY USG
Tomorrow's accuracy is here today in the new U. S. Supergauge. Get complete information about this superb instrument now.
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DIVISION OF AMERICAN MACHINE AND METALS, INC., SELLERSVILLE, PENNA.
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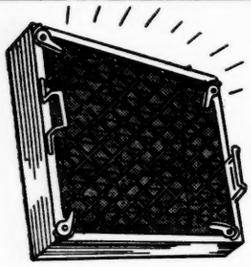

Easier Way to Clean Air Filters

Efficiency of air conditioning units is largely dependent on periodic cleaning of air filters . . . regular, complete removal of deposits entrapped from the air is necessary to maintain high filtering capacity.

If your filters are of the dry or viscous impingement washable type, you can clean them easily and effectively by using Oakite materials specially designed for this work. Dirt, dust, soot, lint, pollen and other foreign matter collected on surfaces are thoroughly removed . . . full filtering capacity is quickly restored . . . equipment out-of-service time is reduced to a minimum.

Dependable Oakite materials and techniques are available for expediting other essential air conditioning maintenance jobs. Write us concerning your problems. There's no obligation.

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OAKITE Specialized CLEANING
MATERIALS · METHODS · SERVICE · FOR EVERY CLEANING REQUIREMENT

ABSTRACTS AND REVIEWS

SMOKE ABATEMENT

All the papers and discussions presented at the 1946 Conference of the Smoke Prevention Association are now available in book form—Proceedings of the 1946 Smoke Prevention Conference. It includes a series of papers on methods of controlling combustion to aid in smoke prevention.

Proceedings of the 1946 Smoke Prevention Conference, leatherette bound, 8½ x 11 in. Published by John Paul Taylor, St. Joseph, Mo. Price $2.

● ● ●

ANALINE—A 14-page pocket size booklet, No. 5 in a series on Controlling Chemical Hazards has been issued by the U. S. Department of Labor to cover the handling and use of aniline in industry. It covers exposure, protective clothing, safety measures and first aid. For copies at 5c each write to the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.

HEAT PUMP HEAT SOURCES—In a Survey of Heat Pump Heat Sources made for the Southeastern Electric Exchange by Southern Research Institute, Birmingham, Ala., Dr. E. N. Kemler, head of the institute reviews the various sources of heat and what has been done in the field of research. He presents information on the use of air, ground water, earth, and other sources as heat reservoirs for the heat pump. A review of the problem indicates that studies must be conducted to determine the heat transfer of long pipes when buried in the ground, and temperature distribution around buried pipes. Report contains 10 graphs and a table giving thermal conductivities of soil and related materials.

HOUSING PRACTICES—National Housing Bulletin 5 issued by the National Housing Agency on Housing Practices—War and Prewar. This 58-page booklet reviews problems of design and construction. It covers such important subjects as changes in construction—materials and structure, mechanical and electrical systems, problems in building codes; trends in dwelling design and site planning—private residential design and planning, and public housing design and planning. For sale by Superintendent of Documents, Government Printing Office, Washington 25, D. C. Price, 15c.

HEAT PUMP INSTALLATIONS—Sabert Oglesby, Jr., has prepared for the Southeastern Electric Exchange a Review of Commercial Heat Pump Installations. It lists 20 installations, and describes 12 of these systems in detail in a report issued by the Southern Research Institute, Birmingham, Ala. Report contains test and operating data.

WHAT READERS SAY

Location of Air Filters

EDITOR, HEATING AND VENTILATING:

We have read with great interest in your October issue the article entitled "Schoolroom Unit Ventilators," by J. E. York. This equipment and the controls seem very well conceived but we, as air filter manufacturers, note a somewhat faulty placement of the filters. We believe that most of the filter manufacturers will agree with us on these points because the writer has discussed a great number of times in meetings the points which we believe are faulty in use of filters in regard to ventilation or heating in such designs.

The filter is placed close over the heating element. It does not protect the heating element and a great amount of dirt or lint which is in the air has to pass through the heating element first before it reaches the filter. What will happen to the dirt or lint when it passes through the heating element? If such dirt or lint passes the heating element without any effect, such as setting up some odors or sticking to the heating element, etc., the filter will most probably catch it. Now the Air Filter Institute recommends that filters should not be operated at a temperature above 150F. It seems that in the design described in the article the temperature might rise above 150F. The reason that the AFI has decided to make such recommendation is to have a safe limit—a safe limit not so much in respect to the filters, but because filter manufacturers know that lint and other organic, and some inorganic, matter caught by the filter may cause odor or other irritating factors. I personally feel that the location, and especially the proximity of the filter to the heating element, is unfortunate.

Now where should the filter be located? There are two ways of locating the filter. One is to locate the filter at the exhaust. This would be preferable from the standpoint of filter temperature application, but would not overcome some of the objections mentioned and in addition such location would not protect the blower motor. The proper location of filters would be, in my opinion, at the wall intake and the recirculating entrance, or room air grille. This last step assures clean outside and clean recirculated air. It will protect equipment, as well as prove the most satisfactory solution from the standpoint of air filtration or clean, sanitary air.

M. H. Kliefoth
Research Products Corporation
Madison, Wis.

NEW CATALOGS

Pressure Reducing Valves

Bulletin 461, 20 pages in two colors, presents engineering, operating and maintenance data on Leslie pressure reducing valves, with capacity tables, descriptions, etc.—*Leslie Co., Lyndhurst, N. J.* 152

for • SPRAYING
• WASHING
• RINSING
• COOLING
• AIR CONDITIONING

YARWAY

SPRAY NOZZLES

FAN-TYPE FOR FLAT SLICING SPRAY

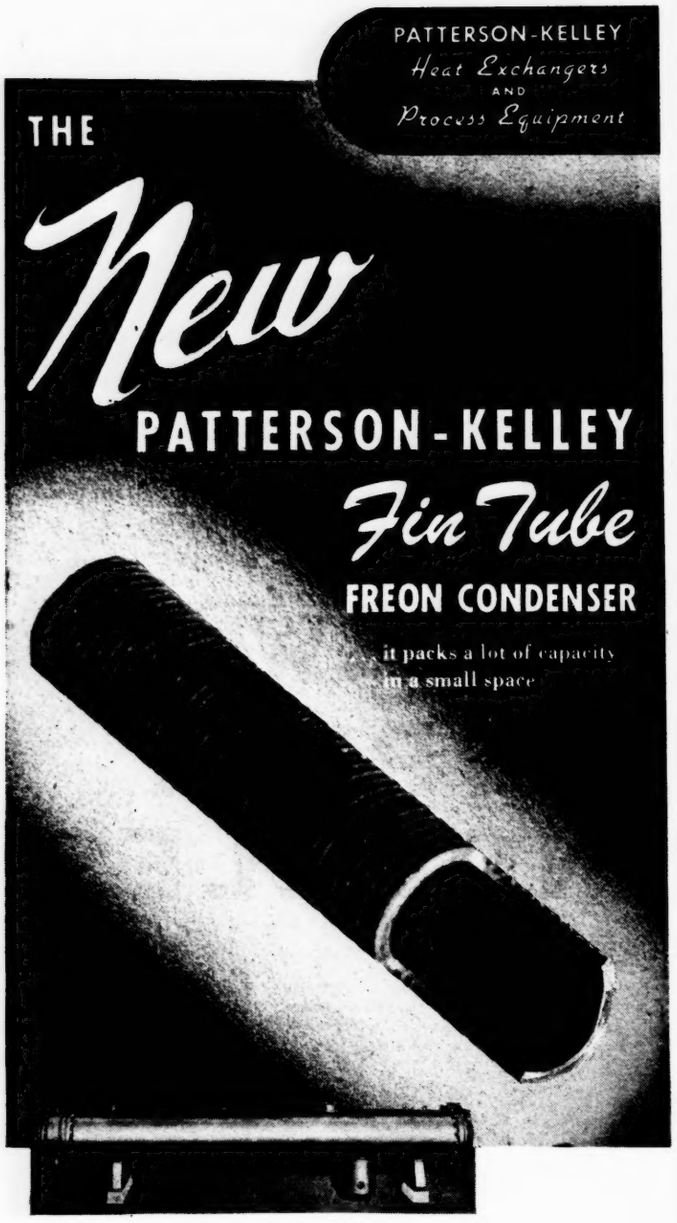
Get the most out of your Spraying Equipment with minimum power . . . with efficient spraying.

Use Yarway Nozzles. No internal vanes or other restrictions to clog or hinder flow. Two types—Yarway Involute-type producing a fine hollow spray with minimum energy loss, and Yarway Fan-type producing a flat fan-shaped spray with time-saving slicing action for cleaning.

Wide range of standard sizes and capacities. Cast or machined from solid bar stock.

Thousands in use. Write for new Bulletin N-616.

YARNALL-WARING COMPANY
104 Mermaid Avenue Philadelphia 18, Pa.



With the fin type of copper tube, approximately 50% extra capacity is provided for any given shell size and tube arrangement over prime surface tube construction. Thus, the buyer of this latest design Patterson-Kelley Freon Condenser can figure on less building space and lower investment cost for his requirements than if he orders a prime surface tube condenser.

For many years, Patterson-Kelley engineers have been specializing in the design and construction of heat transfer equipment, including all types of Freon and other cooling systems. Perhaps with our engineering experience and manufacturing facilities, we can be of service to you in connection with Condensers, Coolers, Heat Exchangers, and other units for your air conditioning or refrigeration systems.

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Representatives in Principal Cities

Stainless Steel Welding Fittings

A 20-page illustrated catalog describes the Tube Turns line of stainless steel welding fittings and flanges. In addition it contains a large amount of engineering data regarding these fittings, the allowable working pressure at various temperatures and for various pipe sizes; internal pressure stress and expansion stress range for austenitic chromium-nickel steels; corrosion resistance data for a large variety of reagents at various temperatures. It contains an article on welding procedure, and data on dimensional tolerances and ASA standards.—*Tube Turns, Inc., Louisville 1, Ky.* **153**

Control Equipment

Complete tab-indexed catalog of control equipment, bound loose-leaf style in simulated leather, presents illustrated descriptions of hydraulic-action controls for various applications; thermostats, fan controls, oil burner controls, stoker controls, relays, industrial oven controls, hot water, steam and refrigeration controls, damper motors, gas valves, and safety panels. The catalog presents data including specifications, dimensions, applications, installation, setting, adjustment and wiring diagrams. — *White-Rodgers Electric Co., 1209 Cass Ave., St. Louis 6, Mo.* **154**

Ceramic Property Chart

The physical, chemical, and electrical properties of AlSiMag ceramics are set forth in a wall chart distributed by the American Lava Corporation. AlSiMag is the trade name of a large family of technical ceramic compositions used as insulators, gas burner tips, flame nozzle tips, extrusion heads, etc.—*American Lava Corp., Chattanooga 5, Tenn.* **155**

High Pressure Hose

A 4-page folder, catalog section 4030, describes high pressure hydraulic control hose for flexible connections to conduct fluids at extremely high pressure as on hydraulic rams and modern pressure greasing equipment.—*B. F. Goodrich Co., Akron, Ohio.* **156**

Sewage Pumps

Bulletin 964-D, 32 pages, 2 colors, describes and illustrates the Buffalo line of non-clogging raw sewage pumps and gives construction details, dimensions, and installation and operating data. All pumps and ejectors are for general service and also for special liquors. —*Buffalo Pumps, Inc., Buffalo 5, N. Y.* **157**

Thermometers and Pressure Gages

Catalog 6707, 20 pages in two colors, presents the Brown line of rectangular gas thermometers and pressure gages. Vapor, gas and mercury actuated thermometers and pressure gages are described with illustrations featuring design improvements.—*The Brown Instrument Co., Philadelphia 44, Pa.* **158**

Refrigeration Service Tools

Catalog No. 46-R, 20 pages, illustrated, presents the Bonney line of refrigeration mechanics' hand tools and sets for refrigeration servicing and repair work.—*Bonney Forge & Tool Works, Allentown, Pa.* **159**

Electric Radiators

An 8-page, two-color booklet covers the Morton Co-Z-Air portable electric radiator. Information includes design features, table of operating costs, and instructions to users.—*Henry J. Morton Associates, Inc., 510 Boulevard Bldg., Detroit 2, Mich.* **160**

Control Equipment

General catalog No. C-4, a loose-leaf bound collection of bulletins describing control equipment for water works, sewage disposal and industry. Products include sequence controls, circuit controls, pump controls, etc. — *Automatic Control Co., 1005 University Ave., Saint Paul 4, Minn.* **161**

Controls

A new catalog describes the various types of automatic and manual re-set thermostats included in the Klixon line. Dimension data, specifications, ratings and other pertinent information is included.—*Spencer Thermostat Company, Attleboro, Mass.* **162**

Steam Condensers

Catalog No. 1461, 43 pages, 3 colors, describes and illustrates a line of steam condensers, steam domes, water boxes and pumping equipment, together with design data, tables, and graphs.—*C. H. Wheeler Manufacturing Co., Lehigh and Sedgley Avenues, Philadelphia 32, Pa.* **163**

Warm Air Panel Heating

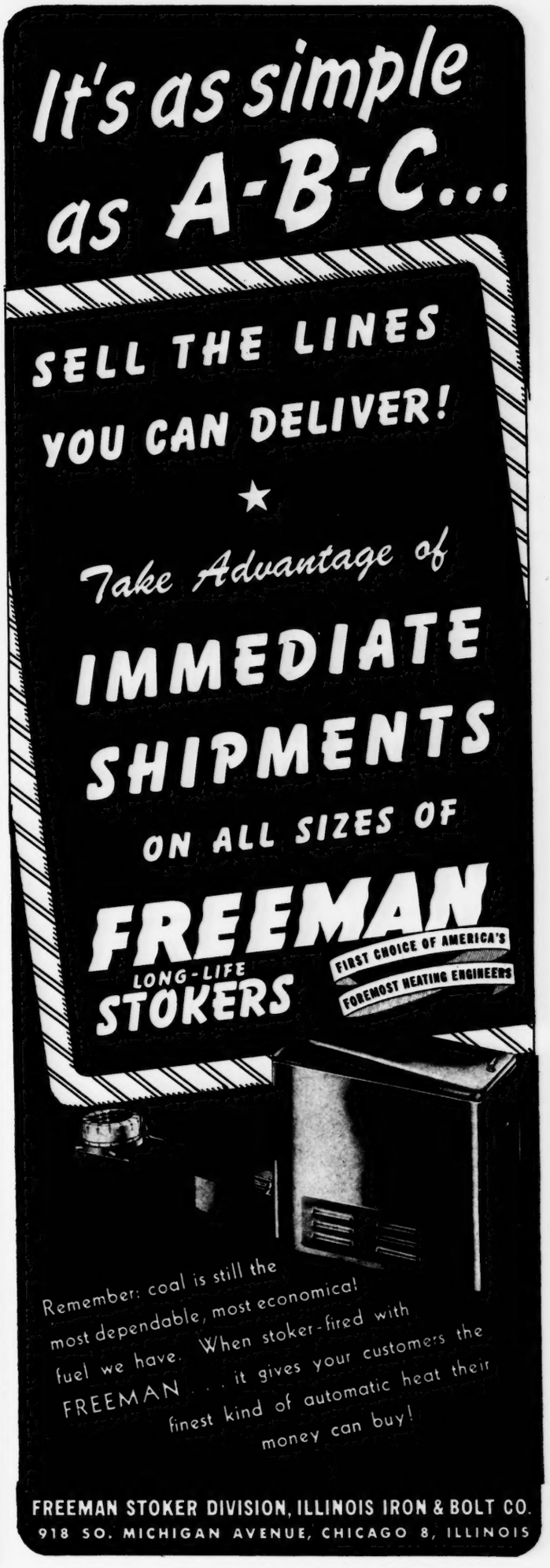
A 14-page brochure presents a series of installation photographs showing the progress and construction of a typical Panelaire system using a Janitrol FAC gas fired heater. — *Surface Combustion Corporation, Toledo 1, Ohio.* **164**

Boilers

Double-pass firebox boilers are described and illustrated in Bulletin No. RM-1, an 8-page folder in two colors. These are riveted or welded, double pass, steel firebox boilers ranging in capacity from 26 to 305 hp. Boilers may be stoker, oil or gas fired.—*The Brownell Company, 410 N. Findlay St., Dayton 1, Ohio.* **165**

Pipe Fabrication

A 12-page booklet describes various pipe and tube coils and special shapes and the methods of their manufacture. Presentation is in the form of an illustrated trip through a fabricating plant.—*Joseph E. Lewis & Co., Inc., 1303-11 Carroll Street, Baltimore 30, Maryland.* **166**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ON ALL SIZES OF
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LONG-LIFE STOKERS
FIRST CHOICE OF AMERICA'S FOREMOST HEATING ENGINEERS
Remember: coal is still the most dependable, most economical fuel we have. When stoker-fired with FREEMAN . . . it gives your customers the finest kind of automatic heat their money can buy!
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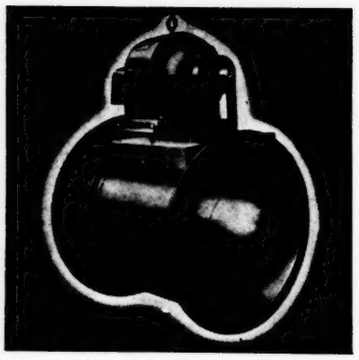

Shown above is the Hartzell Belt-drive Vaneaxial Blower. Available also in Direct-drive models. 18 to 42 inches.

MORE AIR PER DOLLAR AGAINST PRESSURE

How much can Hartzell save for you in air delivery costs? Reduce power consumption just 5 horsepower and over a year of 300 8-hour days at 1½ cents K.W.H. you save $165.00. Certified Test Code figures will open your eyes to the proved economy of air delivery against high pressures afforded by the Hartzell Vaneaxial Blower. Repeated tests show a peak total efficiency of 87%. Quiet and economical against pressures of 8" water gauge and more. Superior performance of one Hartzell Vaneaxial Blower prompted a leading manufacturer to buy 106 more! Get all the facts. Write for Bulletin 1601 today!

Hartzell

PROPELLER FAN CO.

PIQUA DIV. OF CASTLE HILLS CORP. **OHIO**

PEERLESS

VERTICAL,

CLOSE-COUPLED

TURBINE PUMPS

PEERLESS PUMPS

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Tube Expanders

A series of boiler tube expanders is described in a single sheet folder which gives tables of sizes and specifications for use.—*Richard Dudgeon, Inc., 24 Columbia St., New York 2, N. Y.* **167**

Compensators

Bulletin COM-1 presents an explanation of compensators and their various uses in industry, gives data on compensator working pressures and standard end fittings.—*Chicago Metal Hose Corp., Maywood, Ill.* **168**

Steam Generators

A 12-page brochure illustrates the Steam-Pak steam generator featuring low and high pressure lines of steam generators in the range from 12 to 100 horsepower.—*York-Shipley, Inc., York, Pa.* **169**

Vertical Pumps

A new 20-page bulletin, B-159, "Peerless Vertical Close Coupled Turbine Pumps," describes the specialized application of this type of pump to industrial, municipal, engineering, and agricultural installations where short pump settings are encountered and where there is need for deep well turbine pump power and capacity.—*Peerless Pump Div., Food Machinery Corp., 301 West Avenue 26, Los Angeles 31, Calif.* **170**

Pipe and Fitting Materials

Specific information of interest to those using welding fittings and flanges. Chart of pipe and fitting materials lists the composition of the various carbon steels, intermediate alloy steels, stainless steels, and other alloys. It shows for each of the metals listed the service temperature limits, the type welding rod to use, and data on preheat and stress relief for that metal.—*Tube Turns, Inc., Louisville 1, Ky.* **171**

DECEMBER, 1946

TO OBTAIN COPIES OF TRADE LITERATURE listed in this issue, circle on the list below the publications wanted, using the item number at the end of each review.

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. . . Since the Last Issue

B. F. Sturtevant Co., Division of the Westinghouse Electric Corp., has announced the appointment of **Henry F. Daley** as Middle Atlantic district application engineer in Philadelphia. A veteran of 28 years' service with Sturtevant, Mr. Daley held a position as power and marine sales specialist before his newest promotion.

Sturtevant also has announced the appointment of **Eugene N. Foss, 2nd,** as manager of the company's Northwestern district, succeeding Fred Herlan who will retire soon. A great grandson of B. F. Sturtevant, the company's founder, Mr. Foss joined Sturtevant at Hyde Park, Mass., in 1934 after his graduation from Harvard College where he studied mechanical engineering. He was transferred to Washington, D. C. in 1936 as a sales engineer and later was appointed manager. In 1939 he returned to Hyde Park to become assistant manager of the Cooling and Air Conditioning Division, and a year later was appointed assistant to the president.

Promotion of **O. W. Bynum** as manager of direct sales, **John A. Gazelle** as manager of distributor-dealer sales and **Carl U. Spriggs** as assistant general sales manager has been announced by **Carrier Corp.** The appointments were effective November 1. All three will make their headquarters in Syracuse.

A new professional committee on chemistry and toxicology headed by **Henry F. Smyth, Jr.,** Ph.D., senior fellow of Mellon Institute, Pittsburgh, has been announced by the **Industrial Hygiene Foundation.** Another committee, that of preventive engineering, remains under the chairmanship of **Professor Philip Drinker** of the Harvard School of Public Health, but has been assigned several new members. Under a new plan of organization each of four professional committees will be served by a member of the foundation's full time staff acting as secretary. Other committees are the medical, for which **C. Richard Walmer, M.D.,** will serve as secretary, and the legal committee with **John F. McMahon,** foundation managing director, is the acting secretary.

Howard Coonley, chairman of the executive committee of the **American Standards Association** has been elected president of the new International Organization for Standardization, the formation of which has recently been completed by delegates from 25 nations meeting in London. Headquarters will be set up shortly in Geneva, Switzerland. Formation of the new ISO consolidates into a single organization the work of the old International Federation of National Standardizing Associations (ISA) and that of the war-born United National Standards Coordinating Committee. The International Electrotechnical Commission, a third important standardizing agency, is expected to affiliate with ISO shortly as its electrical division.

Captain Thomas H. Urdahl, USNR, and **Commander John Everetts, Jr.,** USNR, both recently released to inactive duty by the Navy, have formed a partnership for the practice of engineering. The firm of Urdahl and Everetts will specialize in problems associated with air conditioning, industrial drying and humidification. An office in charge of Mr. Everetts has been opened at 212 Mission St., San Francisco, and a temporary office in charge of Mr. Urdahl at 726 Jackson Pl., N. W., Washington, D. C.

W. W. Watson, 2970 West Grand Blvd., Room 402, Detroit, Michigan, has been appointed as representative for **H. B. Smith** boilers in the Detroit, Michigan territory.

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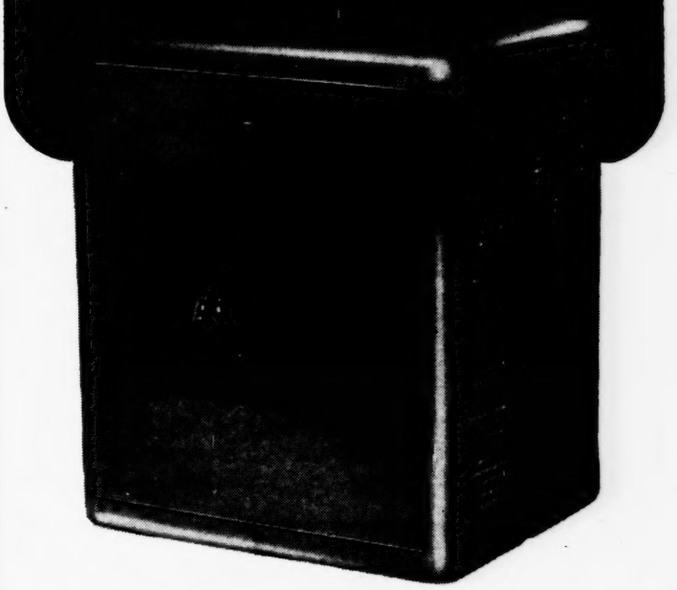

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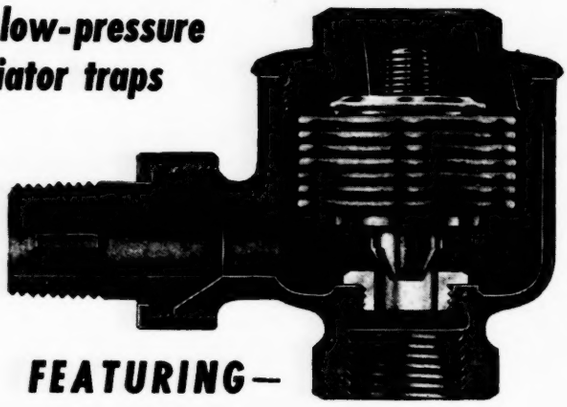

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Middle Atlantic District sales headquarters of the **B. F. Sturtevant Co. Division of Westinghouse Electric** have been moved from Camden to new offices in the Westinghouse Building, 3001 Walnut Street, Philadelphia. The offices were formerly located at the company's plant at Cressmont and Haddon Avenues, Camden.

The Railway Express Agency purchased 500 high-speed steel refrigerator cars of extremely advanced design with the signing of a $7,500,000 contract with the **American Car and Foundry Co.** Although the new cars are known as refrigerator cars, they can be quickly adapted to carry merchandise express traffic and can be equipped with auxiliary hot water heaters to protect perishables in freezing weather.

James W. Speer is serving as district manager of the office that **Cargocair Engineering Corp.** has opened at 726 Jackson Place, N. W., Washington, D. C.

William S. Major, development engineer for **Bituminous Coal Research, Inc.**, Pittsburgh, Pa., was the principal speaker at a meeting of the Solid Fuel Institute in which modern methods of smoke control were discussed. The BCR engineer's talk gave the performance story of overfire jets under fluctuating steam loads.

A series of informative dinner meetings, aimed at acquainting employes of anthracite producing companies with industry problems and methods being used to solve them, will be held in Wilkes-Barre by the **Anthracite Institute** for 5,000 key mining employees. Colliery superintendents, mine foremen, section foremen, outside foremen, fire bosses, department heads and the president, secretary and members of the mine committee of each local of the United Mine Workers of America will be invited to the meetings. It was estimated that it would take the greater part of the winter to complete the full schedule of meetings planned.

Getting Personal

A. G. Dixon *(Will Radiant Heating Displace Convection Heating?*—page 59) is an engineering graduate from the University of Illinois, class of 1924. He studied heating and ventilating under Professor A. C. Willard and served in the U. S. Marines in World War I in France and Germany. He joined Modine as a heating engineer in May, 1926, became assistant sales manager in 1928 and sales manager in 1931 of Modine's Heating and Air Conditioning Division. In 1943 he was elected secretary of the company. He has served on various advisory committees of NRA, OPM, WPB and CPA relative to unit heaters, convectors and other extended surface heating equipment. He is a member of the joint convector code committee which has drafted a code for testing and rating both cast iron and non-ferrous convectors, which code is now in process of becoming a commercial standard of the National Bureau of Standards. Mr. Dixon has been a member of the ASHVE since 1928. He is married and has two children.

The symposium on the Heat Pump, beginning on page 67, presents the results of highly diversified experience on the part of the four authors. **Eugene R. Ambrose** is with a public utility organization. He is a 1930 graduate of

Carnegie Tech. After receiving his degree he completed the General Electric test course, then entered the G.E. refrigeration production department as assistant to the supervisor in charge of testing. He became design engineer of ventilating and air conditioning equipment in 1933, later was made application engineer. Included in his G.E. experience was considerable work on the heat pump.

Mr. Ambrose in 1939 became a member of the engineering staff of American Gas and Electric Service Corporation which has properties in a number of Middle Atlantic and East North Central states. He is engaged in designing, preparing specifications, and supervising installations of heating, ventilating and air conditioning systems for these properties, and these installations include a number of heat pump systems.

A lengthy progress report on the heat pump appeared in the January 1944 issue of H & V under the joint authorship of Philip Sporn, executive vice-president and chief engineer of American Gas and Electric, and Mr. Ambrose. Later these same authors presented a comprehensive discussion of the heat pump defrosting problem; this appeared in the July 1945 issue of H. & V. Since then Messrs. Sporn and Ambrose have been active in proposing an organized study of the heat pump by electric utilities.

E. N. Kemler holds engineering degrees from University of Kansas and a doctor's degree from Pitt. He has been with Dodge Manufacturing Co., instructor and assistant professor at University of Pittsburgh, research engineer for Gulf Oil, and production engineer for Gulf's Gypsy Dursion at Tulsa. He is an authority on fluid flow, piping, and vibration problems. From 1940 to 1945 he was professor of mechanical engineering at Purdue, where he became interested in the heat pump. In 1945 he was made head of the engineering research division of Southern Research Institute, Birmingham, Alabama. One of the projects under his direction there is a study of the heat pump for southeastern electric utilities.

W. L. Holladay was born in Fayette, Mo. He received his B.S. in E.E. in 1924 from the California Institute of Technology. This was followed by a General Electric test course.

During 1925-1927 he introduced the G. E. refrigerator in Texas, Oklahoma and Southern California.

For the next four years he was manager, Product and Engineering Department, The George Belsey Co., Ltd., Los Angeles. From 1942-44 he served as manager of the Central Repair Department, Montgomery Ward and Co., Oakland, Calif. From 1944-1946, he was purchasing agent for the General Air Conditioning and Heating Co., San Francisco.

Mr. Holladay joined Drayer-Hanson in 1946 and has been active in the development and introduction of Airtopia units.

Besides being the author of several papers on refrigeration, he is a member of the ASRE, AIEE and the ASHVE.

INDUSTRIAL DEGREE-DAYS
September and October, 1946

City	Number of Industrial Degree Days			
	55F Base		45F Base	
	September	October	September	October
Baltimore, Md.	0	9	0	0
Buffalo, N. Y.	10	73	0	4
Chicago, Ill.	3	32	0	1
Cleveland, Ohio	3	43	0	0
Detroit, Mich.	7	49	0	0
Indianapolis, Ind.	3	18	0	0
New York, N. Y.	0	12	0	0
Philadelphia, Pa.	0	9	0	0
Pittsburgh, Pa.	3	33	0	0
St. Louis, Mo.	0	13	0	0

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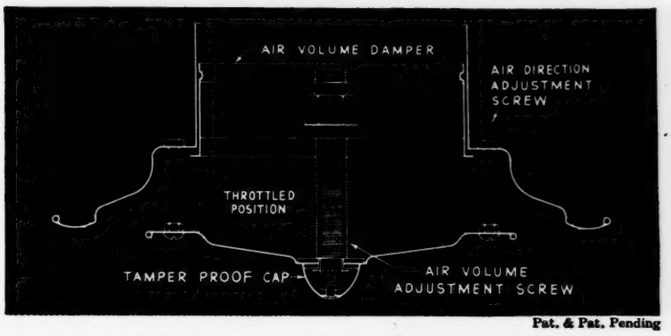

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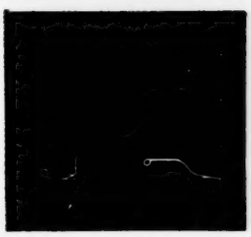

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Canadian Degree-Days for September and October, 1946*

City	September 1946	September Normal	October 1946	October Normal
Calgary, Alta.	411	432	843	722
Charlottetown, P. E. I.	162	222	446	539
Crescent Valley, B. C.	383	360	741	698
Edmonton, Alta.	453	444	849	747
Fort William, Ont.	424	354	676	722
Grand Prairie, Alta.	429	480	843	812
Halifax, N. S.	146	189	411	499
London, Ont.	148	126	338	508
Medicine Hat, Alta.	327	267	794	601
Moncton, N. B.	190	282	490	595
Montreal, P. Q.	138	180	457	561
North Bay, Ont.	270	264	527	594
Ottawa, Ont.	174	204	490	595
Penticton, B. C.	205	213	620	527
Porquis Junction, Ont.	353	396	626	800
Prince George, B. C.	468	444	846	741
Quebec City, P. Q.	181	276	477	651
Regina, Sask.	403	408	896	794
St. John, N. B.	171	270	446	558
Saskatoon, Sask.	400	396	899	781
Toronto, Ont.	124	168	342	504
Vancouver, B. C.	225	234	546	459
Victoria, B. C.	229	264	480	446
Windsor, Ont.	92	54	245	425
Winnipeg, Man.	377	330	794	744

*These data are supplied through the courtesy of the Meteorological Division, Air Services Branch, Department of Transport, Canada, with the cooperation of N. W. Etter, Heating Engineer, Enamel & Heating Products Limited, Sackville, N. B.

COMING EVENTS

DECEMBER 15-18, 1946 — Annual Meeting of American Society of Refrigerating Engineers, Hotel Pennsylvania, New York, N. Y. Secretary, David L. Fiske, 40 W. 40th Street, New York 18, N. Y.

JANUARY 14-17, 1947 — Materials Handling Exposition, Public Auditorium, Cleveland, Ohio. Exposition management, Clapp & Poliak, Inc., 37 Wall Street, New York, N. Y.

JANUARY 27-31, 1947—7th International Heating and Ventilating Exposition, Lakeside Hall, Cleveland, Ohio. Sponsored by The American Society of Heating and Ventilating Engineers in conjunction with their 53rd Annual Meeting. Charles F. Roth, manager, International Exposition Co., Grand Central Palace, New York 17, N. Y.

MARCH 24-29, 1947—Annual Convention, Oil Heat Institute of America, at Hotel Traymore, Atlantic City, N. J. For details write the institute at 30 Rockefeller Plaza, New York 20, N. Y.

MARCH 31—APRIL 2, 1947 — Midwest Power Conference, Palmer House, Chicago. Professor Stanton E. Winston, Illinois Institute of Technology, Chicago 16, Illinois.

APRIL 7-10, 1947—Annual Convention of The National Association of Corrosion Engineers, Palmer House, Chicago, Ill. For details write to Elton Sterrett, Executive Secretary, 905 Southern Standard Building, Houston 2, Texas.

HEAT LOSS THROUGH FLOORS OF BASEMENTLESS BUILDINGS

Recent research by the National Bureau of Standards has resulted in more accurate methods of estimating heat loss through floor slabs laid on the ground, and verified the accuracy of existing data on losses through floors laid over crawl spaces. The data on this sheet are based on the methods proposed by the Bureau.

Floors on Ground

Heat loss through floors laid on the ground can be estimated (for the design, or maximum load, hour) by use of the following formula:

$$Q = LF\ (T_i - T_g)$$

where

Q = heat loss through floor, Btu per hour,
L = length of floor edge adjacent to exposed wall of building, feet,
F = heat loss factor, Btu per degree temperature difference (from Table 1),
T_i = inside design temperature, F,
T_g = average outside temperature for week preceding instant for which estimate is to be made, F.

The last term (T_g) is the only one for which data are not readily available. However, T_g can be defined as the average outside temperature F, for the week preceding the coldest temperature of record.

Such data are simple to compile and are presented in Table 1.

The following example will indicate the application of the equation:

Example. A 30 x 28 ft. residence near New York City is built on a slab of 4 inches of concrete laid on 5 inches of gravel. What will be the floor heat loss?

Solution. The slab is similar to floor type 2 (Table 1), for which $F = 0.69$. The exposed edge L will add up to $(30 + 30 + 28 + 28) = 116$ ft. Presumably the inside temperature would be 70F, and

TABLE 1.—VALUES OF "F" FOR FLOORS LAID ON GROUND

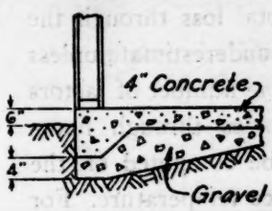
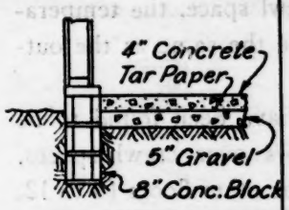
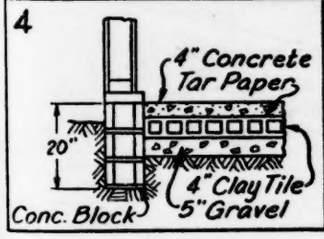
TYPE OF FLOOR	VALUE OF "F" (in Btu per hr. per foot of edge per F)
1 	0.81
2 	0.69
3	0.55
4 	0.75

TABLE 2.—VALUES OF T_g FOR 54 CITIES FOR ESTIMATING HEAT LOSSES THROUGH FLOORS

Albany, N. Y.	5	Milwaukee, Wisc.	1
Atlanta, Ga.	20	Minneapolis, Minn.	—4
Baltimore, Md.	21	Nashville, Tenn.	14
Birmingham, Ala.	19	New Haven, Conn.	9
Boston, Mass.	8	New Orleans, La.	32
Buffalo, N. Y.	6	New York, N. Y.	14
Chicago, Ill.	5	Oklahoma City, Okla.	9
Cincinnati, Ohio	17	Philadelphia, Pa.	18
Cleveland, Ohio	16	Pittsburgh, Pa.	12
Columbus, Ohio	13	Portland, Me.	12
Dallas, Tex.	26	Portland, Ore.	20
Denver, Colo.	8	Rochester, N. Y.	6
Des Moines, Ia.	0	St. Louis, Mo.	15
Detroit, Mich.	14	Salt Lake City, Utah	9
Duluth, Minn.	5	San Diego, Calif.	49
Fort Wayne, Ind.	13	San Francisco, Calif.	45
Grand Rapids, Mich.	8	Savannah, Ga.	38
Harrisburg, Pa.	9	Scranton, Pa.	8
Hartford, Conn.	16	Seattle, Wash.	29
Houston, Tex.	43	Spokane, Wash.	—1
Indianapolis, Ind.	16	Syracuse, N. Y.	9
Kansas City, Mo.	—2	Toledo, Ohio	14
Lincoln, Neb.	0	Topeka, Kansas	—2
Little Rock, Ark.	13	Trenton, N. J.	12
Los Angeles, Calif.	51	Utica, N. Y.	4
Louisville, Ky.	25	Washington, D. C.	14
Memphis, Tenn.	12	Wichita, Kansas	0

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TABLE 3.—VALUES OF "U" FOR FLOORS OVER CRAWL SPACES

TYPE OF FLOOR	VALUE OF "U" (in Btu per hr. per foot of edge per F)
5 Double Wood Flooring, Joist	0.24
6 4" Concrete, 1" Nailing Strips	0.46
7 Carpet, 4" Concrete, 1" Nailing Strips	0.34
8 4" Concrete, 1" Fiberboard	0.30
9 Single Y.P. Flooring, Joist	0.45
10 Single Floor, ½" Rigid Insulation	0.27
11 H.W. Flooring, Subfloor	0.34
12 Battleship Linoleum, Y. P. Floor	0.34

from Table 2, T_g for New York = 14F, so that

$$Q = 116 \times 0.69\ (70 - 14) = 4480 \text{ Btu per hour.}$$

The data are probably on a sufficiently accurate basis for most estimating purposes because the floor heat loss is likely to be small compared to other losses. However, the data are incomplete in that they do not cover the cases of frozen ground and of snow-blanketed ground.

Tests indicate that insulating the floor at the edge is beneficial both in saving heat and in reducing lateral temperature gradients across the floor.

Floors Over Crawl Space

Floors over crawl spaces are illustrated in types 5 to 8, inclusive, Table 3. This type of floor has a conventional heat loss coefficient (air-to-air).

Tests on floors laid over crawl spaces indicated that the factors contained in handbooks are suitable for estimating heat losses through such floors, except in the case of a floor which is heavily insulated on the underside. In this case, the edge loss increases largely in comparison to the total loss through the floor, and this may result in an underestimate unless it is taken into consideration. The number of factors involved indicate that heat losses through floors laid over crawl spaces should be computed on the basis of an estimated crawl-space temperature. For a continuously ventilated crawl space, the temperature should be assumed to be the same as the outdoor temperature.

Since the usual U values have been found to be reasonably accurate for floors over crawl spaces, Table 3 gives U values for four other floors (9 to 12, inclusive) *not tested* by the Bureau. These values are from H & V's Reference Data 73-74, and are included here for convenience.

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HEAT GAINS AND LOSSES FROM AIR DUCTS

The purpose of the accompanying charts is to determine (1) heat losses or gains from ducts and (2) the temperature of the air leaving the duct. The charts were designed by the late Ensign Charles H. Kilpatrick while a student at the University of California.

The chart is designed for use with circular ducts but can be used for rectangular ducts as explained later. The results are for a duct length of 100 ft.

Example. Determine the temperature rise of air passing through and the heat gained by a duct 4.1 ft in diameter and 100 ft long insulated with a material 0.8 in. thick having a conductivity of 0.40 Btu per hr per sq ft per inch per F, carrying air entering the duct at 40F, moving through the duct at 2200 fpm, with air surrounding the duct at 90F. Density of air at 40F = 0.07 lb per cu ft.

Solution. Enter Fig. 1 at the lower left scale for duct diameter (4.1). Move vertically to the curve for 2200 fpm velocity. Move horizontally to the right to the curve located as follows: On the lower right scale for insulation thickness locate 0.8 inches, move vertically to the conductivity

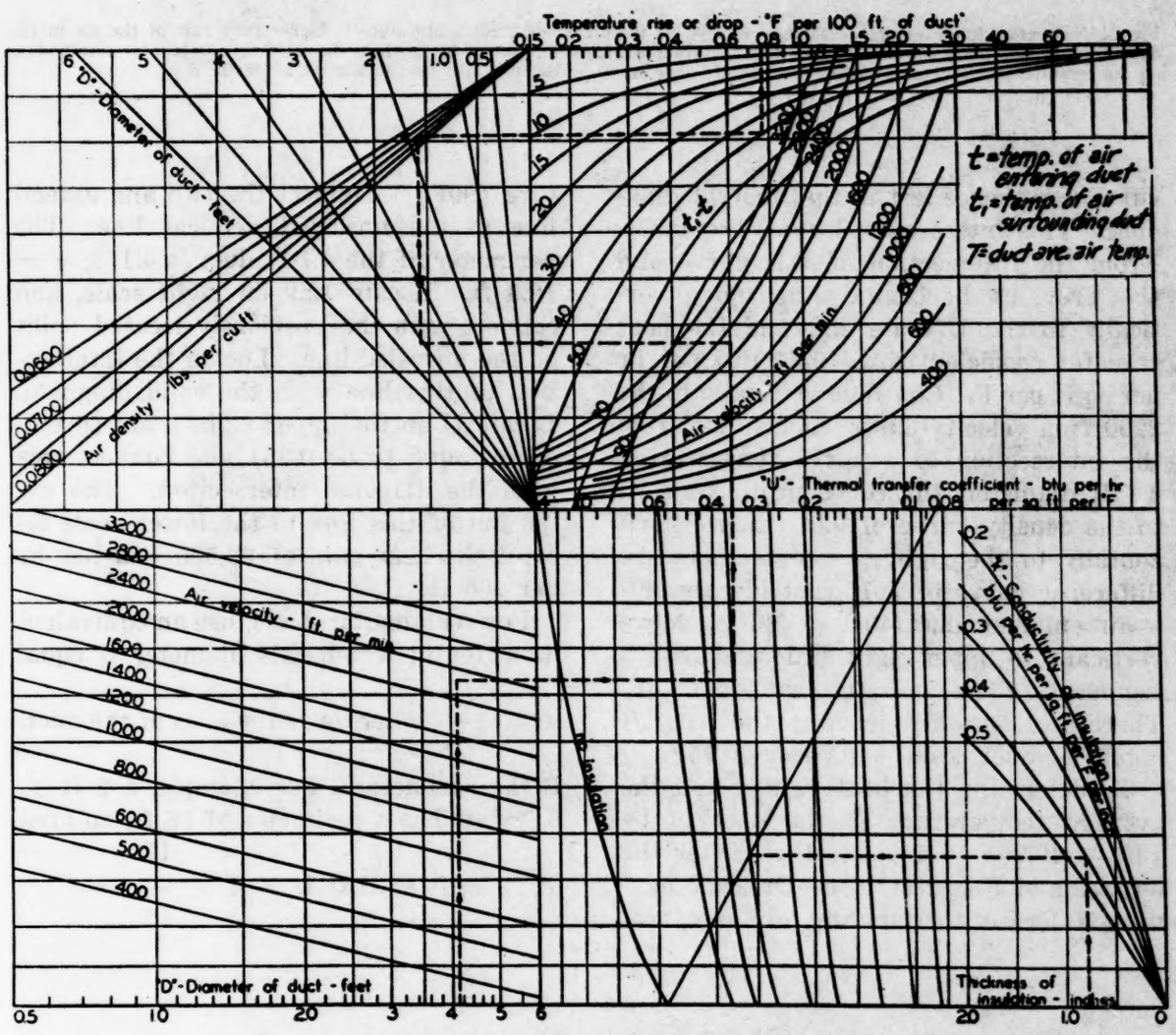
Temperature rise or drop - °F per 100 ft. of duct
"D" - Diameter of duct - feet
Air density - lbs. per cu ft
Air velocity - ft. per min
t = temp. of air entering duct
t₁ = temp. of air surrounding duct
T = duct ave. air temp.
"U" - Thermal transfer coefficient - btu per hr per sq ft per °F
K - Conductivity of insulation - btu per hr per sq ft per °F per inch
Thickness of insulation - inches
"D" - Diameter of duct - feet

Fig. 1. Graph for determining U and temperature rise or drop.

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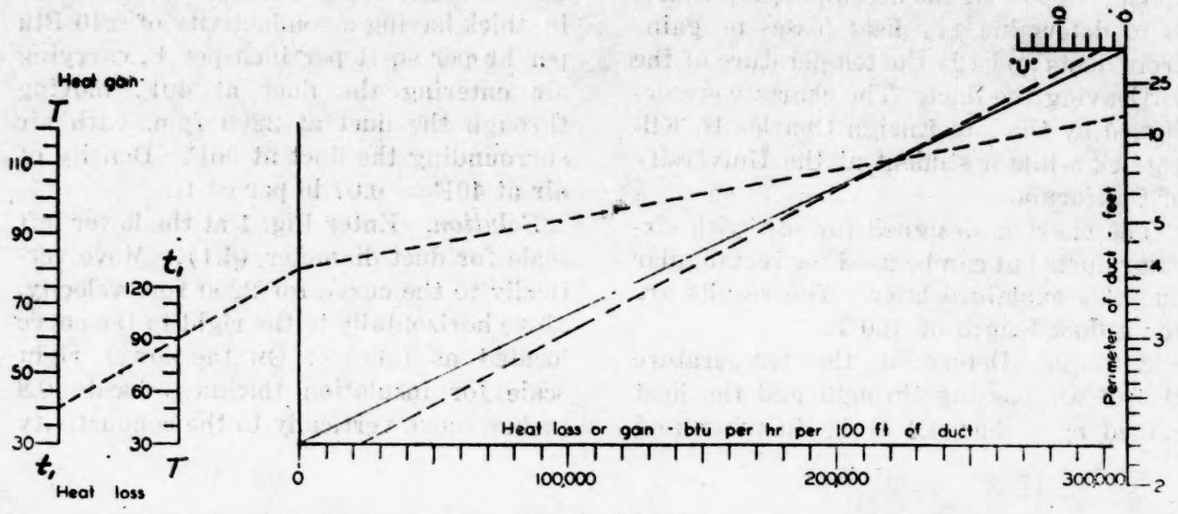

Fig. 2. Nomogram for determining heat gain or loss. If the surrounding air temperature is higher than that of the air in the duct, there is heat gain, and use the left scale for average temperature of air in the duct, the scale next to left for surrounding air temperature. In the case of heat loss, the left scale is for surrounding air, next to left for air in the duct.

curve 0.4, thence left to the straight diagonal. This intersection locates the curve. From the intersection of this curve and the previous horizontal line move vertically to the U scale and find the heat transfer coefficient to be 0.35 Btu per hr per sq ft per F. Continue vertically to the 2200 fpm velocity curve, thence left from the intersection to a curve representing 4.1 ft diameter, thence vertically upward to the density curve of 0.07, thence horizontally to the right to the temperature difference curve of 50F (outside air 90F —air entering duct 40F = 50F). Move vertically to upper right scale and read a temperature rise of the air of 0.75F. Therefore, the air leaving the *100 ft length* of duct is 40 + 0.75 = 40.75F.

To determine the heat gained find the average temperature in the duct to be (40 + 40.75) ÷ 2 = 40.4F. Enter the left scale of Fig. 2 at 40.4. On scale next to left locate surrounding air temperature (90F). Connect the two and extend line to the unscaled vertical line. The perimeter of the 4.1 ft duct is $4.1 \times \pi =$ 12.9 ft. Locate 12.9 on right scale, and connect with the previously located point on the unscaled line. Locate the intersection of this line with the solid diagonal. Locate U on the upper right scale (previously found to be 0.35) and connect this with the diagonal intersection. The extension of this line to the lower scale locates the heat gain of 22,500 Btu per hr per 100 ft.

For rectangular ducts, use an equivalent diameter in which this diameter is equal to $4 \times \frac{A}{P}$, where A is the area of the duct, P the perimeter. For example, a 2 ft × 6 ft duct has a perimeter of 16 ft, an area of 12 sq ft so that $D = 4 \times \frac{12}{16} = 3$.

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CHARACTERISTICS OF AIR CLEANING METHODS

The accompanying tables summarize the characteristics of various air cleaning methods while the curve shows the size distribution of outdoor and industrial dusts. All these data are by J. M. Dalla Valle, consulting engineer.

The size distributions shown on the graph are a composite of a large number of determinations. They are, however, thoroughly representative and show the size range of particles ordinarily dealt with in collection problems.

The distributions for both outdoor and industrial dusts are for stable conditions—that is for situations where settling has eliminated most particles above 1.5 microns in diameter. Turbulence may, under some conditions, tend to sustain particles of larger diameter, but unless ducts leading to the air cleaners are near a source of dust, the distributions shown in the figure give substantially the range of sizes which must be captured. The importance of size distribution in testing the

CHARACTERISTICS OF VARIOUS AIR CLEANING METHODS

Device	Description	Application	Rates of Flow, Lin. FPM*	Initial Resistance, Inches of Water	Efficiency, Percent
Cloth	Any cloth or felt medium stretched in suitable frame, so mounted that dust-laden air must pass through it	Industrial general ventilation. Mostly used for collection of valuable product	0.5 to 10	0.05 to 0.2	60 to 90 (ASHVE). General Vent., 85 to 99. (Weight basis, industrial collection)
Paper and Glass Fibre	Any air pervious paper or glass wool stretched and mounted as above. Glass fibre may be in sheets or in bulk evenly distributed	General ventilation although industrial uses not uncommon	20 to 80	0.01 to 0.2	60 to 90 (ASHVE) 20 to 50 (USBS)
Metal (Stationary and Automatic Self-Cleaning)	Metal mesh or eschelon grid. Usually from 1 to 4 in. thick and coated with viscous material	General and industrial ventilation	250 to 300	0.05 to 0.1	60 to 90 (ASHVE)
Electrostatic (Stationary and Automatic Self-Cleaning)	Series of positive and negative plates parallel to air flow. Preionizing wires usually provided. Plates coated with viscous material	General ventilation	300 to 500	0.01	80 to 90 (U. S. Bureau of Standards discoloration)
Water Spray	Chamber with water spray arrangement so that all air moves through a barrier of fine mist and droplets	General ventilation	—	0.05	50 to 60 (ASHVE)
Scrubbers	Device so designed that air is brought into close contact with water. Some forms agitate water mechanically	Cleaning gases also in some forms used for industrial ventilation	For industrial work. Rated at about 1000 cfm per foot length	4 to 8	70 to 90 (Weight basis)
Cyclone	Particles separated by centrifugal action of vortically moving air steam	Industrial dust collection	Entrance velocities from 2000 to 4000	2 to 4	40 to 80 (Weight basis)
Dynamic Fan-Collector	Usually a fan with blade and housing design such that particles are removed by centrifugal action. May be operated wet	Industrial dust collection	Tip blade speeds 7000 to 10,000	About twice loss through fan of same capacity	50 to 90 (Weight basis)

*Rates of flow for filter media are based on extended surface and not on frame size

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CHARACTERISTICS OF AIR CLEANING METHODS

CHARACTERISTICS OF VARIOUS AIR CLEANING METHODS

Method of Cleaning	Initial Cost per CFM, Cents**	Upkeep per CFM-Year, Cents***	Remarks
Shaking at specified time intervals or at fixed pressure drop	20 to 60	5 to 10	In collection of valuable product upkeep may exceed amounts given, especially if material collected is corrosive.
Replacement	2 to 5	1 to 2	In large installations paper bulk is appreciable. Not efficient for particles much below 1 micron.
Washing	1 to 2	Labor for cleaning filters (except automatic types); must be washed every few days.	For efficient collection filters must be kept clean and coated with viscous medium.
Washing, unless self-cleaning	20 to 50	Labor for cleaning and replacing preionizing wires	Units require attention for efficient performance.
Draining	20 to 30 for static types 15 to 20 for mechanical types	Low	Not very efficient, as dust collector nozzles require frequent attention. Units require large horse power.
Material removed from hopper	3 to 5	Low	Not efficient for fine particles unless carefully designed.
Material removed from hopper	20 to 40 (Without motor)	Low for dry types. Add cost of water for wet types	Combination of prime mover and collector saves on space. Not efficient for fine particles.

**Does not cover cost of installation.

***Does not include labor costs. Figures as given are based on 10 hours per day operation, but it is obvious that dust load affects costs materially.

efficiency of dust collection is recognized in the Bureau of Standards procedure. Other methods generally emphasize the qualitative aspects of the test dust.

Some estimate of the effect of settling or suspended dust is obtained from the following tabulation of dust having a specific gravity of 1.0:

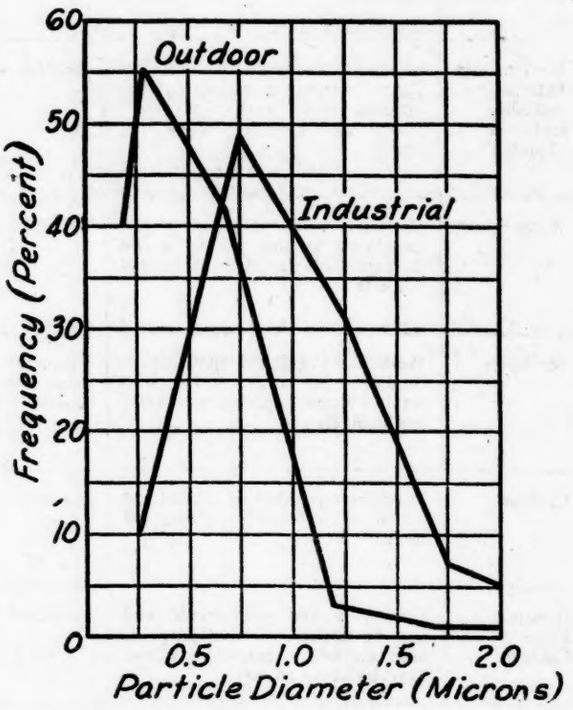
10 micron dust settles at the rate of 13000 ft per hr.

1.0 micron dust settles at the rate of 130 ft per hr.

0.1 micron dust settles at the rate of 1.3 ft per hr.

These rates are based on Stokes' law. The settling rate for particles having specific gravities greater or less than 1.0 are obtained by multiplying the values on the right by the specific gravity.

Air cleaners used for recirculating efficient air cannot be rated by ordinary commercial tests since most States require that recirculated air contain less than a stipulated number of dust particles. Thus far there are no standard methods available for conducting efficiency tests of this sort.



Particle-size distributions of outdoor dust in a typical city and in an industrial plant.

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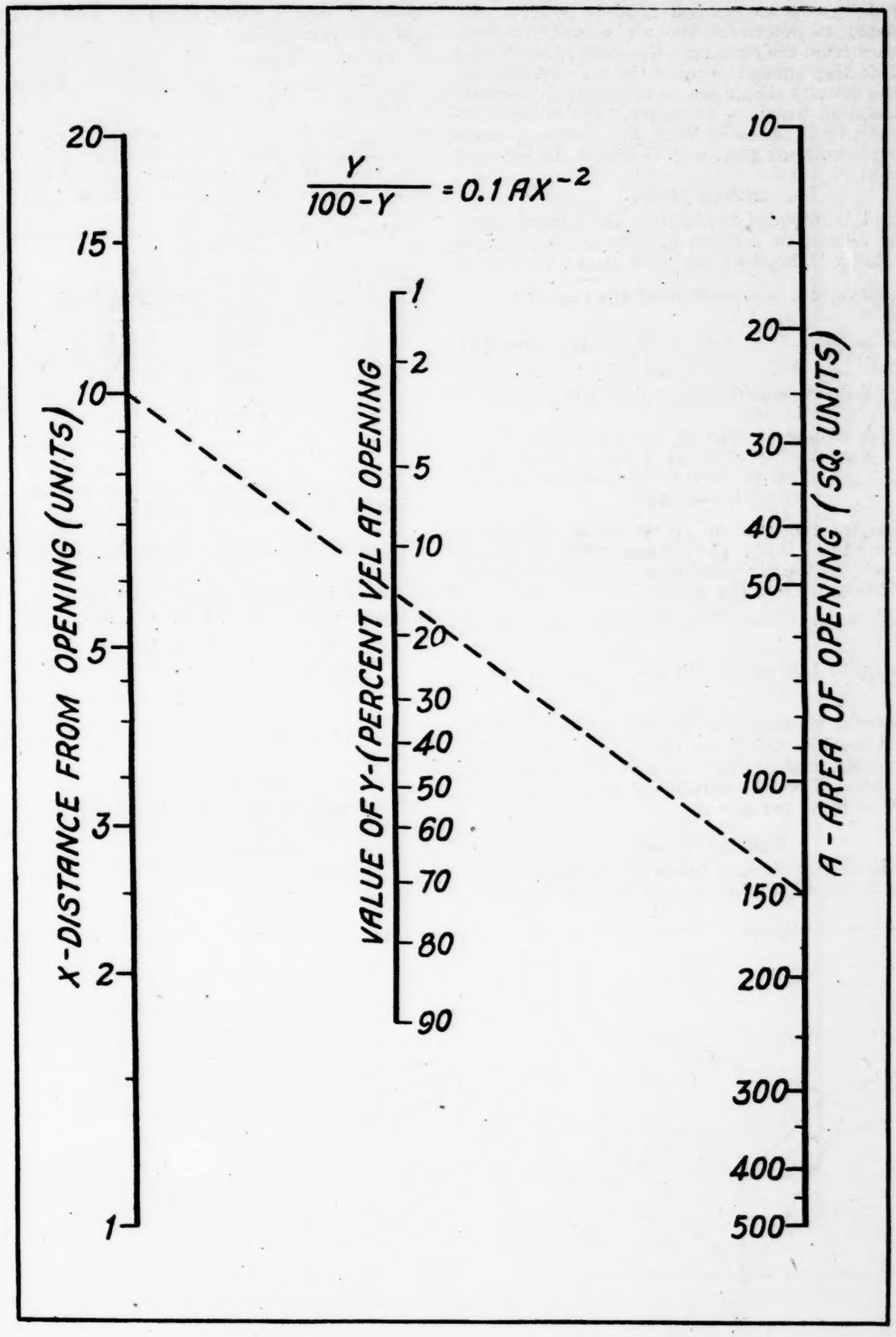

Fig. 1. Nomogram for solution of the equation $\frac{Y}{100-Y} = \frac{0.1A}{X^2}$.

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AIR VELOCITIES NEAR EXHAUST HOODS

In designing an exhaust hood it is frequently necessary to determine the air velocity at some distance from the opening. Generally, the velocity must be high enough to carry the air contaminant, but the velocity should not be too high, since power consumption would be excessive. The accompanying data, by J. M. Dalla Valle, are useful in determining conditions necessary to obtain the required velocity.

Simple Hoods

Fig. 1 is intended to assist in the determination of the velocity at a given distance from a simple (not flanged) hood of the type shown in Fig. 2.

This nomogram is a solution of the equation

$$\frac{Y}{100-Y} = \frac{0.1A}{X^2} \qquad \text{Equation (1)}$$

where X = distance from opening, along axis, inches;
A = area of opening, square inches;
Y = velocity at point X inches from opening in percent of average velocity through opening.

Example. 416 cfm of air is to be exhausted through the 10 in. × 15 in. hood shown in Fig. 2. What will be the air velocity at point S 10 inches along the axis from the hood?

Solution. A 10 × 15 in. hood has an area of 150 sq in. or 1.04 sq ft. Then $\frac{416}{1.04} = 400$ ft per min average velocity over the face of the opening. Then A = 150 and X = 10. Lay a rule across these points in the outer scales of Fig. 1 and find Y = 15%. Then the velocity at point S = 15% × 400 or 60 ft per min.

Flanged Hoods

In the case of flanged hoods of the type shown in Fig. 3, the edge effect is such that the equation given in Fig. 1 does not apply. The equation for flanged openings is

$$\frac{Y}{100-Y} = \frac{cA^{0.82}}{X^{1.5}} \qquad \text{Equation (2)}$$

where the symbols are identical with those in Equation (1), and c is a variable depending on the ratio of the dimensions of the rectangular hood. Values of c are given in Table 1.

Table 1. Values of c for use in Equation (2)

Ratio of smaller dimension of hood opening to that of larger hood opening	Value of c	Ratio of smaller dimension of hood opening to that of larger hood opening	Value of c
0.1	0.046	0.6	0.126
0.2	0.072	0.7	0.132
0.3	0.090	0.8	0.138
0.4	0.105	0.9	0.142
0.5	0.117	1.0	0.144

Example. Assuming the hood in Fig. 2 and the foregoing example were flanged as in Fig. 3, what would be the velocity at point S?

Solution. The ratio of hood dimensions is $\frac{10}{15}$ or .67, so that from Table 1, the value of c would be about 0.130. Then, from Equation (2),

$$\frac{Y}{100-Y} = \frac{0.130 \times 150^{0.82}}{10^{1.5}} = 0.54$$

$$Y = 35\%$$

so that the velocity at S is 35% × 400 = 140 ft per min.

Circular Openings

For circular hoods, both simple and flanged, use Equation (3)

$$\frac{Y}{100-Y} = \frac{0.0825A^{1.04}}{X^{1.91}} \qquad \text{Equation (3)}$$

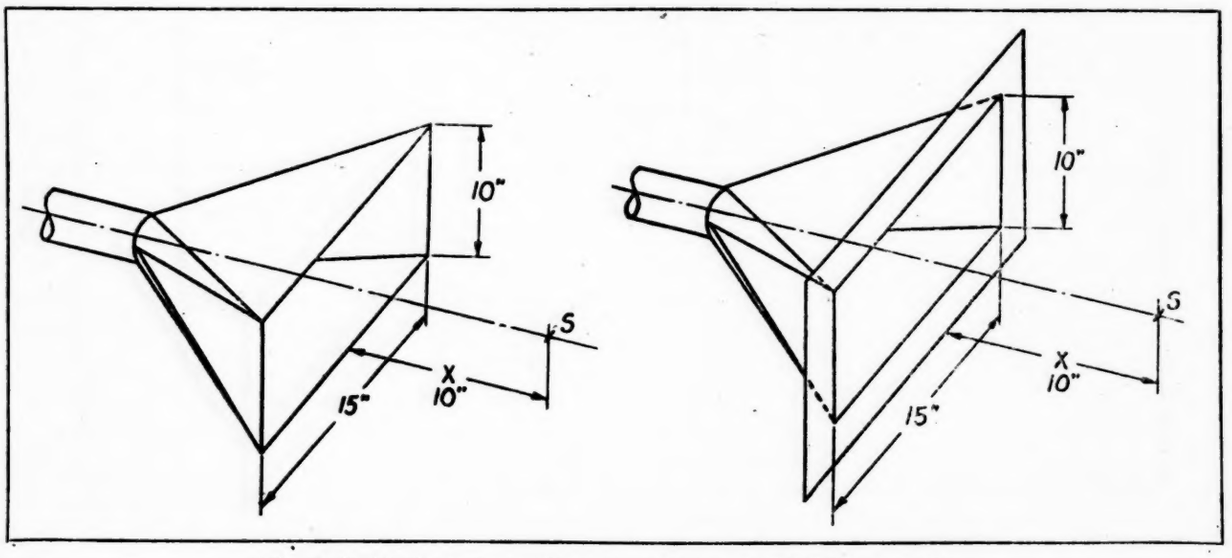

Fig. 2. (Left) Simple rectangular hood of the type to which Fig. 1 applies.
Fig. 3. (Right) Flanged rectangular hood of the type to which Equation (2) applies.

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HEAT EMISSION BY RADIATION

TABLE 1.—RADIATION FROM SURFACES TO SURROUNDINGS AT ABSOLUTE ZERO
(When emissivity = 1.0)

Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. Per Hr.	Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. Per Hr.	Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. Per Hr.	Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. Per Hr.
30	99.3	84	150.8	57	123.0	111	183.3
31	100.0	85	151.9	58	124.0	112	184.7
32	101.0	86	153.1	59	125.1	113	186.0
33	101.7	87	154.1	60	125.9	114	186.9
34	102.3	88	155.4	61	126.9	115	188.5
35	103.3	89	156.5	62	127.7	116	189.9
36	104.2	90	157.7	63	128.8	117	191.2
37	105.0	91	159.0	64	130.0	118	192.5
38	105.9	92	160.0	65	130.8	119	193.9
39	106.7	93	161.2	66	131.8	120	194.6
40	107.6	94	162.4	67	132.8	121	195.9
41	108.3	95	163.5	68	133.9	122	197.3
42	109.3	96	164.6	69	134.8	123	198.8
43	110.1	97	165.8	70	135.9	124	200.0
44	111.1	98	167.0	71	136.8	125	201.4
45	112.2	99	168.5	72	137.9	126	202.8
46	113.1	100	169.7	73	139.0	127	204.2
47	113.9	101	170.9	74	140.0	128	205.5
48	114.7	102	172.2	75	141.1	129	206.9
49	115.5	103	173.3	76	142.2	130	208.4
50	116.6	104	174.4	77	143.3	135	215.3
51	117.6	105	175.5	78	144.4	140	222.8
52	118.4	106	176.6	79	145.4	145	230.4
53	119.4	107	176.9	80	146.5	150	238.1
54	120.2	108	179.3	81	147.6	175	279.6
55	121.1	109	180.6	82	148.6	200	326.3
56	122.1	110	182.0	83	149.6		

The Stefan-Boltzmann law states that the radiation from a black body to another body is proportional to the difference between the fourth power of the absolute temperatures of the two bodies. When the radiating body is not a perfect black one,

$$Q_r = \frac{0.174\ Ae\ (T_h^4 - T_c^4)}{10^8}$$

where Q_r = radiation in Btu per hour
A = area in square feet
e = emissivity of radiating body
T_h = absolute temperature of radiating body
T_c = absolute temperature of receiving body

When $A = 1$, and when the receiving body is at absolute zero, the equation reduces to

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$$q = \frac{0.174e\,T_h^4}{10^8}$$

Table 1 is a tabular solution of this equation. It is useful in determining the unit radiation by giving the radiation to absolute zero of the hot body, then that to absolute zero by the receiving body and subtracting to find the difference which is the unit radiation when the emissivity is 1.0. Table 2 gives values for *e*.

Example. What is the radiation from a rough plaster ceiling at 100F to surrounding surfaces at a temperature of 50F?

Solution. From Table 1, *q* for the 100F ceiling to absolute zero surroundings is 169.7, that from the 50F surroundings is 116.6, or (169.7 — 116.6) = 53.1 Btu per sq ft per hr at an *e* of 1. From Table 2, *e* for rough plaster is 0.91, so that the radiation is 53.1 × .91 = 48.3 Btu per sq ft per hour.

Fig. 1 is a graphical solution of equation (1) for special conditions, such as with radiant heating, and when *e* = 1.0. Note that neither the table nor the curve includes the heat emitted by convection.

TABLE 2.—EMISSIVITY OF VARIOUS SURFACES (From Hottel)

Asbestos board	0.96
Asbestos paper	0.93-0.945
Brick, red	0.93
Enamel, white, on iron	0.897
Glass, smooth	0.92-0.94
Gypsum	0.903
Marble, light gray, polished	0.903
Oak, planed	0.895
Black lacquer on iron	0.875
Oil paints, all colors	0.92-0.96
Aluminum paint	0.27-0.67
Paper	0.924-0.944
Plaster, rough	0.91
Absolute black surface	1.00
Dark paint	0.92
Rough concrete	0.94
Lime wash	0.91

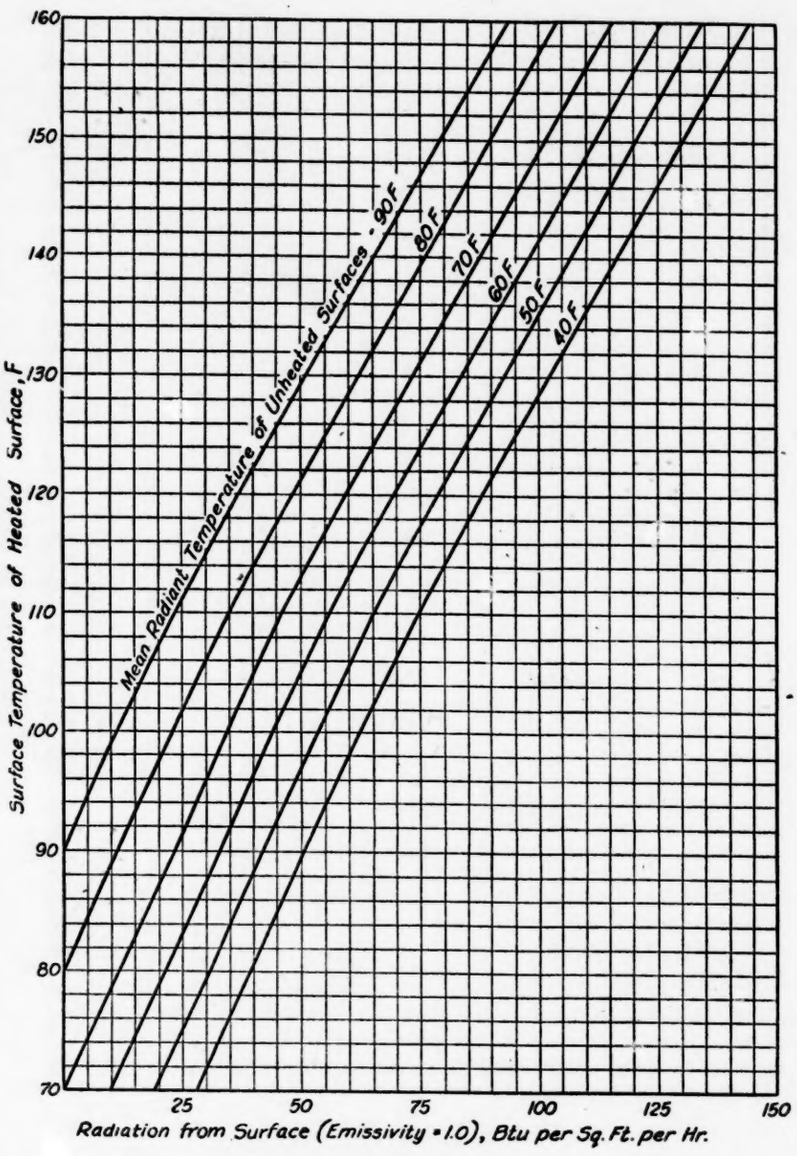

Fig. 1. Heat emission by radiation when the emissivity is equal to 1.0.

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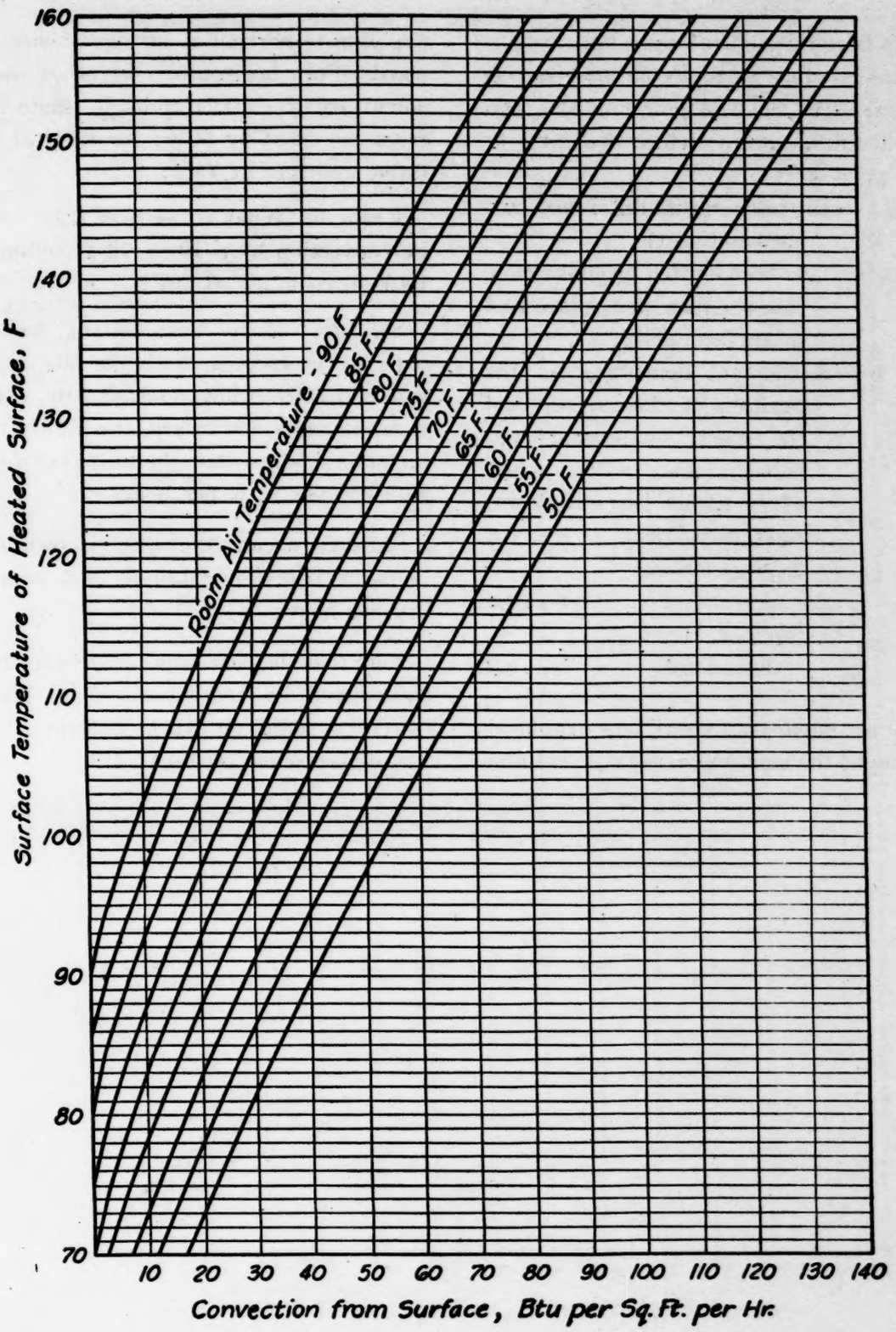
HEAT EMISSION FROM FLAT SURFACES TO AIR BY NATURAL CONVECTION



Values Found on this Scale are for Horizontal Surfaces Facing Upward. For Downward Surfaces, Multiply Chart Value by 0.5; Vertical Surfaces by 0.75.

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HEAT EMISSION FROM FLAT SURFACES TO AIR BY NATURAL CONVECTION

The heat transmitted by a unit area of a heated surface to air by natural convection varies as the 1.25 power of the temperature difference (surface and air), or

$$q_c = bt^{1.25}$$

where t = the temperature difference, surface and air, F

q_c = heat transmitted by convection, Btu per hour per square foot of surface, and

b = a constant depending on the position of the surface and the mean temperature involved, and with values (from Fishenden) as follows:

Surface vertical	b = 0.3
Surface facing upward	b = 0.4
Surface facing downward	b = 0.2

The accompanying chart is a graphical solution of the equation using b = 0.4 and applying to horizontal surfaces facing upward. For horizontal surfaces facing downward (b = 0.2) multiply values read from the chart by 50%; for vertical surfaces, multiply by 75%.

Example. What is the heat transferred by convection by a 10 × 12 ft ceiling at 100F to room air at 50F?

Solution. Move from 100 on the left scale to intersection with the 50F curve, and vertically below read 53 Btu. Since a ceiling faces downward, the multiplying factor is 0.50, so that the unit transfer is 26.5 Btu per sq ft per hour.

Since the area is 120 sq ft, the total convectional transfer is 120 × 26.5 = 3180 Btu per hour.

Note that the data apply only to the heat transferred by convection. Heat transferred by radiation can be determined by use of Reference Data 317-318.

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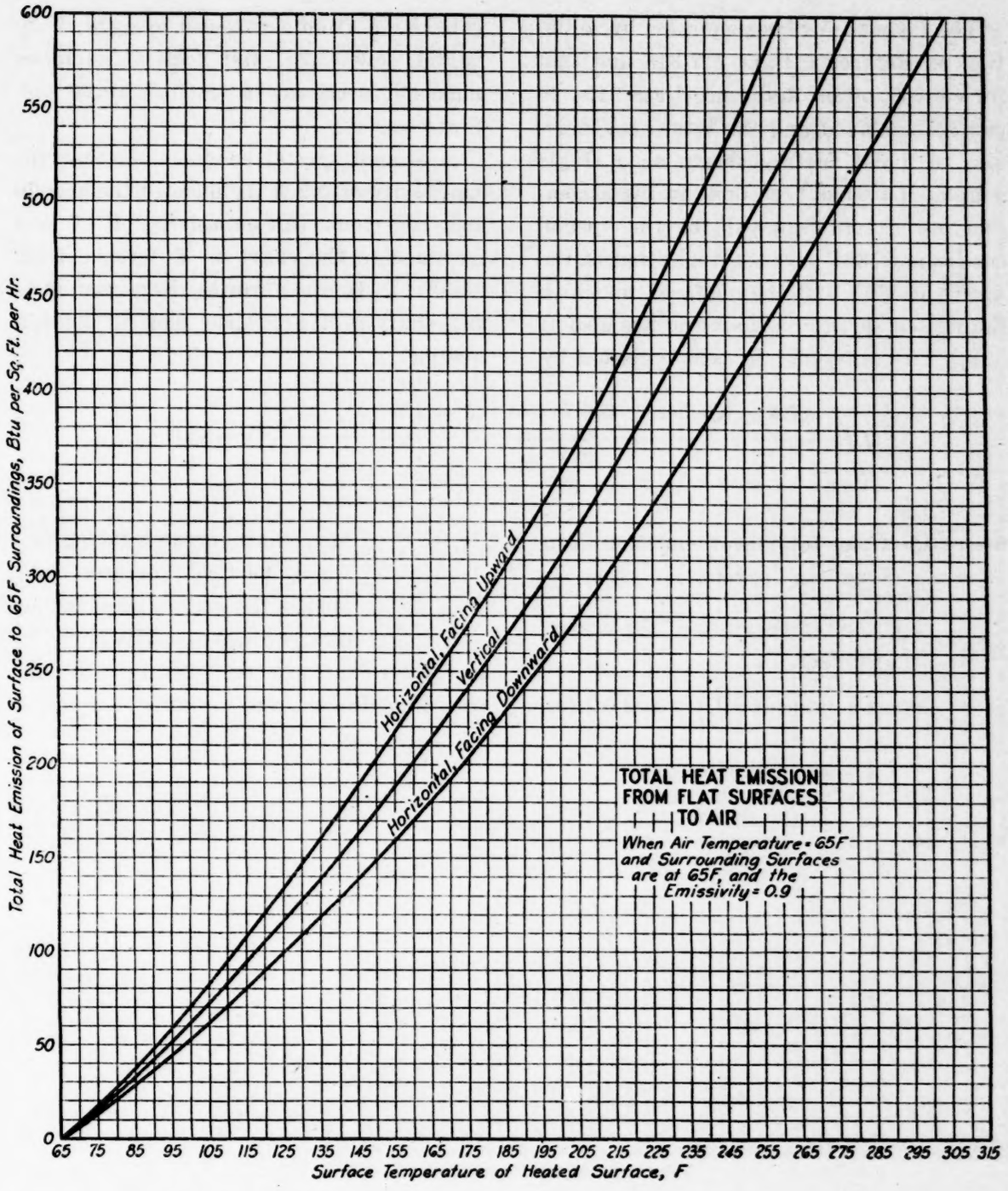
HEAT EMISSION FROM FLAT SURFACES TO AIR BY RADIATION AND CONVECTION


Total Heat Emission of Surface to 65F Surroundings, Btu per Sq. Ft. per Hr.
0
50
100
150
200
250
300
350
400
450
500
550
600
65 75 85 95 105 115 125 135 145 155 165 175 185 195 205 215 225 235 245 255 265 275 285 295 305 315
Surface Temperature of Heated Surface, F
Horizontal, Facing Upward
Vertical
Horizontal, Facing Downward
TOTAL HEAT EMISSION FROM FLAT SURFACES TO AIR
When Air Temperature = 65F and Surrounding Surfaces are at 65F, and the Emissivity = 0.9

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HEAT EMISSION FROM FLAT SURFACES TO AIR BY RADIATION AND CONVECTION

The formula for heat emission by radiation on Reference Data 317-318 and that for convection on Reference Data 319-320 can be combined and the total heat emission of a flat surface shown on a single graph. This has been done in the accompanying chart, designed for the special condition of 65F surroundings; that is, the air is at 65F, and the surfaces to which the heated surface is radiating are also at 65F. The difference in heat emission by convection from horizontal upward, horizontal downward, and vertical surfaces, makes it necessary to include three separate curves.

A second special condition is that the emissivity of 0.9 is included in the accompanying chart (an emissivity of 1.0 was assumed in the chart on Reference Data 317-318). In other words, the accompanying chart is a graphical plot of the following:

$$Q = \frac{17.4 \times 0.9 \left[(t + 459.7)^4 - (65 + 459.7)^4\right]}{10^{10}} + b\,(t - 65)^{1.25}$$

where Q is the total heat emission, Btu per sq ft per hr,

t is the temperature of the heated surface, F,

b has a value of 0.4, 0.3 and 0.2 for horizontal surfaces facing upward, vertical surfaces, and horizontal surfaces facing downward, respectively

and where the air temperature and surrounding surfaces are at 65F.

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DEFINITIONS OF TERMS USED IN HEATING, VENTILATING AND AIR CONDITIONING

It is the purpose of this sheet and one to follow to define some of the terms more or less peculiar to heating, ventilating and air conditioning work, and further, to make these definitions insofar as possible, simple and understandable. With this latter point in mind, the treatment has been somewhat liberalized in that the definition is concerned more with common usage than academic precision.

In compiling this list special attention has been paid to terms for which definitions are not elsewhere easily located, such as Clo, Met, Cold 70, etc.

Absolute pressure. The pressure above an absolute vacuum or, in other words, the sum of the pressure shown by a gage and that indicated by the barometer.

Absorption. An assimilation during which the absorbent undergoes a physical or chemical (or both) change. Absorbers such as lithium chloride and calcium chloride are used in dehumidifying. The phenomenon is also used in absorption refrigeration.

Absorption refrigeration. A process whereby a secondary fluid absorbs the refrigerant, and in doing so gives up heat, then releases the refrigerant, during which it absorbs heat.

Absorptivity. The ratio of the radiant energy absorbed by a body to that falling upon it. It is equal to the emissivity for radiation of the same wave length.

Adiabatic. At constant total heat; an action or process during which no heat is added or subtracted.

Adsorption. A phenomenon consisting of the adhesion, in a very thin layer of gas molecules, of dissolved substances or liquids, to the surfaces of solid bodies with which they are in contact. Silica gel is an adsorber used in dehumidifying while activated carbon is an adsorber used to remove odors from air. During adsorption the adsorber undergoes no permanent physical or chemical change.

Air change. The quantity of infiltration or ventilation air in cubic feet per hour or minute divided by the volume of the room gives the number of so-called air changes during that interval of time, and tables of the recommended number of such air changes for various type rooms are used for estimating purposes.

Air conditioning. The simultaneous production or altering under control of several factors to obtain a desired atmospheric environment. These factors include raising or lowering of temperature, raising or lowering of humidity, distribution of air, and reduction of contaminants, such as dust, fumes, bacteria, odors, etc.

Airfoil fan. A fan with an airfoil-shaped blade which moves the air in the general direction of the axis about which it rotates.

Air washer. A device through which air is passed and washed with water from sprays for the purpose of humidifying, dehumidifying, or cleaning.

Ambient. Encompassing on all sides. Thus ambient air is the air surrounding.

Anthracite. A dense hard coal with a very high percentage of fixed carbon, usually above 90%.

Apparatus dew-point. If the condition line (which see) crosses the saturation line on the psychrometric, the point of intersection is the apparatus dew point.

Aspect ratio. In a duct, the depth of elbow along (parallel to) the axis of bend divided by the width in the plane of the bend.

Atmospheric pressure. Pressure indicated by a barometer. The standard atmospheric pressure is 760 mm. of mercury (29.921 inches of mercury) or 14.696 lb. per sq. in.

Attic fan. A fan so mounted in the attic that it can draw air from the house at night, causing cool outside air to flow into the house, and exhausting the warm air to the outside.

Automatic heating. A central heating system operated without manual attention. Usually means oil fired, gas fired, or stoker fired furnaces and boilers.

Axial flow fan. A fan with a disc or airfoil shaped blade mounted on a shaft and which moves the air in the general direction of the shaft axis.

Basal metabolism. The metabolism of an organism in the fasting or resting state. The rate at which the organism gives off heat at this state is the basal metabolic rate, and this rate is the lowest for the organism under normal conditions.

Baseboard radiator. An arrangement of heating surface located at or replacing the room baseboard, and applied as an alternative to conventional convectors or radiators.

Baudelot cooler. An arrangement of pipes one above another through which refrigerant flows and is vaporized as it absorbs heat from the water being cooled by trickling over the tubes.

Bellows. An expansible metal device containing a fluid which will volatilize at some desired temperature, expand the device and open or close an opening or a switch, as in controls and steam traps.

Bi-metal. Two metals of different coefficients of expansion welded together so that the piece will bend in one direction when heated, in the other when cooled, and thus can be used in opening and closing electrical circuits, as in thermostats.

Bituminous coal. Coal covering a wide range of properties, but in general with a fixed carbon content under 80% and volatile matter of over 20%.

Black body. A body that absorbs all the radiant energy falling upon it.

Blast coil. Heat transfer surface, most frequently of an extended surface arrangement, over which air is blown to be heated or cooled, depending on the temperature of the fluid within the pipelike surface.

Boiler. A closed vessel in which fuel is burned to generate steam or heat water. The boiler may be made either of steel or, as is frequently the case when low pressure steam is used, of cast iron.

Boiler burner unit. Boiler designed especially for gas or oil and sold integrally with the burner.

Boiler heating surface. That part of the interior surface of a boiler subjected to heat on the one side and transmitting the heat to water (or steam or other fluid) within the boiler. Direct surface is that subjected to direct radiant rays of the burning fuel; other heating surface is termed indirect.

Boiler horsepower. The power required to evaporate 34.5 lb. of water at 212F per hour or the equivalent in 33,475 Btu per hr.

Boot. Sheet metal transformation pieces used in warm air heating and connected to horizontal round leaders on one end and to vertical rectangular stacks on the other.

Brine. A secondary liquid cooled by refrigeration and circulated to heat transfer for absorbing heat from air, water or other fluid.

British thermal unit (Btu). The unit of energy or heat required to raise the temperature of one pound of water from 39F to 40F or from 60F to 61F under standard pressure, or 1/180 of the heat to raise 1 lb. from 32F to 212F.

By-pass. An alternative path, in a duct or pipe, for a fluid to flow from one point to another, with the direction determined by the opening or closing of valves or dampers in the main line as well as in the by-pass.

Central fan system. An indirect system of heating in which the air is heated by steam or hot water at a central location and carried to or from the rooms to be heated by a fan and a system of ducts. Also termed Hot Blast System.

Calorie. In engineering, the large calorie is usually used, defined as one one-hundredth of the energy or heat required to raise the temperature of one kilogram of water from 0C to 100C; the small calorie is the heat required similarly to raise the temperature of 1 gram of water.

Central heating plant. Either (a) one heating plant in a building serving all or most of the rooms in the building, as distinguished from individual room heaters, or (b) one heating plant serving two or more buildings. The latter is also termed District heating.

Chimney effect. See Stack effect.

Clo. A unit of clothing insulation defined as the insulation necessary to keep a sitting man comfortable in a normally ventilated room at 70F and 50% relative humidity. In physical terms, it is equal to 42.7 Btu per sq. ft. per hour.

Coefficient of heat transmission. The quantity of heat (usually Btu) transmitted from fluid to fluid per unit of time (usually 1 hour) per unit of surface (usually 1 sq. ft.) through a material or arrangement of materials under a unit temperature differential (usually 1F) between fluids. Commonly used for building materials.

Cold 70. That condition, during the heating season, in spaces where 70° is the desired air temperature, when the air temperature at the thermostat is at 70° but the temperature at the floor or below the thermostat is lower than 70°. This stratification results in relatively long "off" periods of the heating system and a feeling of discomfort even though 70° is maintained at the control.

Comfort chart. A psychrometric chart developed by the American Society of Heating and Ventilating Engineers on which are plotted the zones of comfort for winter and summer, according to the sensation reported by subjects in varying environments.

Condensers. Condensers are used to liquefy a gas. Two commonly used types of condensers for air conditioning work are (1) Shell and Tube which consists of a large number of tubes inside a steel shell or cylinder. Cooling water circulates inside the tubes while vapor flows into the shell, circulates around outside the tubes, is condensed and drawn off the bottom as a liquid; the other is (2) the Evaporative condenser which consists of a coil over which water is sprayed while air is blown across the tubes of the coil. The refrigerant vapor is cooled, or condensed, inside the tubes due to the extraction of heat caused by the evaporation of the water on the outside.

Condensing unit. An assembly attached to one base, and including refrigerating compressor, motor, condenser, receiver, and necessary accessories.

Condition line. The infinite number of combinations of wet and dry bulb temperatures which will satisfy the require-

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DEFINITIONS OF TERMS USED IN HEATING, VENTILATING AND AIR CONDITIONING

ments of an air supply for a given room condition form what is known as the condition line.

Conductance. The quantity of heat (usually Btu) transmitted per unit of time (usually 1 hour) from a unit (usually a square foot) of surface to an opposite unit of surface of material under a unit temperature (usually 1F) differential between the surfaces.

Conduction. The transmission of heat from one part of a body to another part of the same body, or from one body to another in contact with it, without appreciable displacement of the particles of the body.

Conductivity. The quantity of heat (usually Btu) transmitted per unit of time (usually 1 hour) from a unit surface (usually 1 sq. ft.) to an opposite unit of surface of one material per unit of thickness (usually 1 inch, but occasionally 1 foot) under a unit temperature differential (usually 1F) between the surfaces.

Convection. The transfer of heat from one point to another within a fluid by the mixing of one portion of the fluid with another. When the motion is due to differences in density, from temperature differences, the convection is natural; if the motion is imparted mechanically, it is forced convection.

Convector. An arrangement of heat transfer surface contained in an enclosure with openings at top and bottom, thus heating air by convection.

Conversion burners. Fuel-burning devices (usually oil or gas) intended for installation in a wide variety of boilers or furnaces.

Convertor. Usually, a heat exchanger for transferring heat from steam to water.

Decibel. A sound or noise unit, based on the ratio of sound intensity to that at an arbitrary level. One decibel is equal to $10 \log \frac{X_2}{X_1}$ where X_2 is the unknown intensity, X_1, the reference intensity. A bel is equal to $\frac{X_2}{X_1}$ and is thus one tenth decibel.

Degree-day. The product of one day and the number of degrees F the daily mean temperature is below 65F. Thus on a day when the mean temperature is 40F, there are 25 degree-days. The degree-day unit is used in eliminating the weather variable in determining heating load efficiency and in predicting fuel consumption.

Degree-hour. The product of one hour and (usually) the number of degrees F the hourly mean temperature is above a base point, usually 85F. Thus in an hour when the temperature is 92F, there is 7 degree-hours. The degree-hour is used in measuring, roughly, the cooling load in summer for cases where process heat, heat from human beings, and humidity are relatively unimportant as compared with dry bulb temperature.

Degrees Kelvin. Absolute temperature on the Centigrade scale, or degrees C plus 273.16.

Degrees Rankine. Absolute temperature on the Fahrenheit scale, or degrees F plus 459.6.

Dehumidification. The process of removing water vapor (steam) from the air.

Dehydration. The process of removing water from solids or liquids.

Density. The weight of a unit of volume —usually pounds per cubic foot.

Design temperature. The temperature which an apparatus or system is designed to (1) maintain or (2) operate against under most extreme conditions. The former is the inside design temperature, the latter outside design temperature.

Dew-point temperature. The temperature at which a given mixture of air and water vapor is saturated.

Direct expansion. An arrangement of a refrigerant evaporator where the refrigerant itself expands in an evaporator in the air stream.

Disc fan. An axial-flow fan with blades formed of flat sheets or plates so bent that when rotated the blades impart to the air a motion along the axis of the fan shaft.

Direct fired unit. A heater in which the flame's heat is transferred to metal plates and thence directly to the air to be heated. A warm air furnace is direct fired; a boiler is indirect.

Draft. The pressure difference causing flow of a fluid, usually applied to convection flow, such as chimneys, and usually measured in inches of water.

Draft stabilizer. A device to maintain a constant draft in a fuel burning device.

Drip. A pipe connection, usually trapped, between the steam side of a steam piping system and the condensate or return side of the system.

Dry air. Air with no water vapor.

Dry bulb temperature. The temperature measured by a conventional thermometer.

Dry return. A condensate line in a steam heating system carrying both water and air, usually located above the boiler water line.

Duct fan. An axial flow fan mounted in, or intended for mounting in, a section of duct. See Tubeaxial fan and Vaneaxial fan.

Effective temperature. An arbitrary index developed by the American Society of Heating and Ventilating Engineers and combining, in a single value or index number, the effect of temperature, humidity, and air motion on the sensation of comfort. The numerical value is that of the temperature of still, saturated air which induces an identical feeling of comfort.

Efficiency. Ratio of output to input. Frequently misused where efficacy, load factor, or coefficient of performance would better apply.

Ejector. A device in which a high velocity jet acts to entrain mechanically a second fluid to withdraw it from some region of like pressure and to deliver with low turbulence the mixture to a region of higher pressure.

Electronic filter. An air cleaner in which matter in the air stream is electrically charged, then attracted to surfaces oppositely charged.

Emissivity. The ratio of radiant energy emitted by a body to that emitted by a perfect black body. A perfect black body has an emissivity of 1; a perfect reflector an emissivity of 0.

Enthalpy. For most engineering purposes, heat content or total heat, above some base temperature. Specific enthalpy is the ratio of total heat to weight of substance.

Entropy. The ratio of heat added to a substance to the absolute temperature at which the heat is added. Specific entropy is the ratio of entropy to weight of substance.

Equivalent direct radiation (EDR). Heat expressed in terms of a square foot of steam radiator surface emitting 240 Btu per hour; in other words, Btu per hour divided by 240.

Equivalent evaporation. The quantity of water which would be evaporated by a given apparatus if the water is received by the apparatus at 212F and vaporized at that temperature under atmospheric pressure. It is expressed in pounds per hour.

Equivalent length. The resistance of a duct or pipe elbow, valve, damper, orifice, bend, fitting, or other obstruction to flow expressed in the number of feet of straight duct or pipe of the same diameter which would have the same resistance.

Equivalent temperature. Also British equivalent temperature (BET). A composite of MRT (see Mean Radiant Temperature) and air temperature, also defined as the mean temperature of the environment effective in controlling the rate of sensible heat loss from a black body in still air when the surface temperature and size of the black body are comparable to those of the human body. Where the enclosure surface (MRT) and air temperatures are equal, this temperature is also the BET; when not equal the BET is that temperature at which a body with an 80F surface temperature will lose sensible heat at the same rate as in the given environment.

Eupatheoscope. A black hollow copper cylinder heated electrically to the surface temperature of a human being. The heat loss from the cylinder in a given environment is equivalent to that lost by a human being; thus, equivalent temperature, which see.

Evaporative cooling. Cooling by the evaporation of water, heat for which is supplied by the air; feasible where the wet bulb depression is marked, and consequently widely used in dry climates.

Film coefficient. Also called unit convection conductance and surface coefficient. The heat transferred by convection per unit area per degree temperature difference between the surface and the fluid.

Finned tube. Extended (also termed gilled) surface in the form of fins on tubes or pipe.

Flooded system. In refrigeration, a system where the refrigerant passing the evaporator is only partially evaporated.

Foaming. In a boiler, the carry-over of slugs of water into the piping, due to dirty water. Also see Priming.

Fog. Condensed water vapor in very fine particles and lying near the ground, unlike a cloud in this latter respect.

Free area. The total area of open space in a grille through which air can pass.

Fume. Air carrying certain chemical contaminants of very small size ranging from one tenth to one micron in diameter.

Furnace. Either the combustion space in a fuel burning device or a direct fired air heater. In the latter case, not to be confused with a boiler.

Furnace burner unit. Warm air furnace sold integrally with the burner.

Furnace volume. The cubic contents of the combustion space of a boiler bounded by the grates, direct heating surface, and tube sheet.

Fusible plug. An insert of a low melting metal placed in boilers, sprinklers and other devices to melt when the temperature becomes dangerously high, so that the melting will relieve pressure, allow water flow or otherwise tend to alleviate the dangerous condition.

Gage pressure. Pressure above that shown by the barometer.

Globe thermometer. A thermometer in a hollow spherical black globe, readings from which show a higher value, due to radiation, than a conventional thermometer so that the globe device measures the effectual radiation temperature.

Grate area. Area of surface on which fuel is burned, ordinarily measured at the top surface of the grates.

Grille. A covering over an air inlet or outlet with openings through which air passes.

Head. A unit of pressure intensity usually given in inches or feet of a column of the fluid under consideration. Thus, 1 foot of water head is the pressure from a column of water 1 ft. high.

(Concluded on Data Sheet 325-326)

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DEFINITIONS OF TERMS USED IN HEATING, VENTILATING AND AIR CONDITIONING

(Concluded from Data Sheet 323-324)

Heating effect. A term not yet precisely defined but having to do with the ratio of heat output of a radiator or convector in the living zone of a room with the total output measured by steam condensation or water temperature drop.

Heating medium. A fluid used for conveying heat from a heat source to heat dissipating devices; includes air, water and steam.

Heat pump. A mechanical refrigerating system used for air cooling in the summer and which, when the evaporator and condenser effects are reversed, absorbs heat from the outside air (or water) in winter and raises it to a higher potential so that it also can be used for winter heating.

Heating surface. Broadly, the area intended for transferring heat.

Heating systems. Central Fan System, which see.

High side. That portion of a refrigerating system under the relatively low pressure of the evaporator.

Hot water heating system. One in which water is the heating medium.

Humid heat. Ratio of the increase in total heat per pound of dry air to the rise in temperature, with constant pressure and humidity ratio.

Humidity. Usually, water vapor mixed with dry air. Absolute humidity is the weight of water (or steam) per unit volume of the air-water mixture.

Humidity ratio. See specific humidity.

Hydrostatic test. On a boiler, the closing of all openings and pumping water into the boiler at a pressure 50% greater than the normal operating pressure. The purpose is to locate leaks or prove that there are no leaks.

Hygrometer (Also Hygroscope). A device for measuring moisture in the air.

Hygroscopic. Readily absorbing and retaining moisture.

Hygrostat. A device sensitive to humidity changes and arranged to actuate other equipments when a predetermined humidity is attained.

Inch of water. A unit of pressure equivalent to 0.036136 pounds per square inch.

Indirect heating surface. Any type of heating surface which does not come in direct contact with the original source of the heat.

Industrial degree-day. A degree-day (which see) unit based on a (usually) 45F or 55F mean daily temperature so as to be applicable to industrial buildings maintained at relatively low temperatures.

Initial pressure. In a process, the temperature of the fluid entering the process. Final pressure is that of fluid leaving process.

Infiltration. Leaking inward. In heat transmission, applies to the air entering the space through cracks, doors, etc. The opposite of Exfiltration.

Infra-red. Radiant energy of a wave frequency inferior to visible red light, which, when it falls on a body, causes its temperature to rise. Commonly called radiant heat.

Ingot iron. A commercially pure iron, lower in carbon than carbon steel, used chiefly in sheets.

Isothermal. Constant temperature.

Kata thermometer. A thermometer developed by Heberden, Hill, and Griffiths which is heated before being exposed to the environment to be measured and the rate of fall noted. The rate of fall of the dry bulb is a measure of the radiation and convection; of the wet-bulb, a measure of evaporation plus radiation and convection. If the dry bulb is cooled in a vacuum bottle the rate is a measure of radiation alone.

Latent heat. The quantity of heat required to produce a change of state at constant temperature; usual cases are latent heat of vaporization of water (970.2 Btu per lb. for water to steam at 14.696 lb. per sq. in. abs. pressure) and latent heat of fusion (143.6 Btu per lb. for ice to water).

Leader. In warm air heating, the supply duct running horizontally from the furnace to the riser, or stack.

Load factor. Ratio of average output during a period to maximum output during the period. Sometimes, ratio of output to maximum capacity.

Mbh. One thousand Btu per hr.

Mean radiant temperature (MRT). That single temperature of all enclosing surfaces which would result in the same heat emission as the same surfaces with various different temperatures.

Mercury switch. A glass tube employing mercury to establish electrical contact between circuits when the tube is tilted so that the mercury bridges the gap between contacts, and conversely.

Met. A convenient and approximate unit of human heat production equivalent to the average metabolic heat produced by resting man, about 18.50 Btu per sq. ft. per hr.

Metabolism. The chemical change in living cells by which energy is provided for activities and vital processes.

Micron. One thousandth of a millimeter.

Milinch. One thousandth of an inch.

One-pipe hot water system. One in which the heated and cooled water to and from the radiators are carried in the same "main" but in the same direction, with the hot water on top.

One-pipe steam system. A steam heating system in which steam supplied to radiator travels in same pipe but in opposite direction from cold water or condensate returning from radiator.

Operative temperature. Operative temperature is that temperature of an imaginary environment in which, with equal wall (enclosing areas) and ambient air temperatures and some standard rate of air motion, the human body would lose the same amount of heat by radiation and convection as it would in some actual environment at unequal wall and air temperatures and for some other rate of air motion. The term originated at the John B. Pierce Laboratory of Hygiene.

Orifice. Opening. Commonly used to apply to discs placed in pipe lines or radiator valves to reduce the fluid flow to a desired amount.

Orsat. A device for measuring the percentage of carbon dioxide, oxygen and carbon monoxide in flue gases.

Paddle wheel fan. A centrifugal fan with radial blades.

Panel heating. See Radiant heating.

Panel radiator. A radiating surface (see Radiant heating) primarily flat to increase its ability to transmit radiant energy.

pH. Hydrogen ion concentration. The logarithm of the reciprocal of the gram ionic hydrogen equivalents per liter. On the pH scale 7.0 is normal, smaller values are acid, progressively so, while higher values to 14 are alkaline, progressively so.

Pipe coil. Pipe used for heating, usually in (1) parallel lines connected to headers at the end, forming a grid; (2) in parallel lines with ends alternately connected to adjacent lines by bends, forming a so-called continuous coil.

Plenum chamber. A space or chamber, part of and connected to a duct system, under a somewhat higher than atmospheric pressure.

Priming. In a boiler, the excessive carry-over of fine water particles with the steam due to insufficient steam space or faulty boiler design.

Propeller fan. A fan with airfoil blades and which moves the air in the general direction of the axis of the fan.

Psychrometer. A device for measuring the humidity in the air and employing a wet bulb and a dry bulb thermometer.

Psychrometric chart. A graph showing the properties of air steam mixtures and used in air conditioning.

Quality. The ratio by weight of vapor to liquid plus vapor in a mixture, as in steam.

Radiant heating. Usually, a system of heating by surfaces at higher than body temperatures whereby the rate of heat-loss from the human beings by radiation is reduced.

Radiation. The transfer of energy in wave form from a hot body to a (relatively) cold body independent of matter between the two bodies.

Recessed radiator. One set into the wall but not enclosed.

Reflectivity. The ratio of radiant energy reflected by a body to that falling upon it.

Refrigerant. Any substance which will absorb heat while vaporizing.

Relative humidity. Either (a) the ratio of partial pressure of the water vapor in the air to the pressure at saturation, or (b) the ratio of water vapor by weight per unit volume to the weight of saturation.

Resistance. In heat transfer, the reciprocal of Conductance, which see.

Resistivity. The reciprocal of Conductivity, which see.

Return air. Air returning to a heater or conditioner from the heated or conditioned space.

Reverse cycle refrigeration. See Heat pump.

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Reynolds number. A dimensionless ratio proportional to pipe or duct diameter, velocity, and density of fluid and inversely proportional to the viscosity of the fluid. With English units, a Reynolds number over 2500 indicates turbulent flow; below that, streamlined flow.

Saturated air. Air containing saturated water vapor with both air and vapor at the same dry bulb temperature.

Saturated steam. Steam at the boiling temperature corresponding to the pressure at which it exists. Dry saturated steam contains no water particles in suspension. Wet saturated steam does.

Saturation pressure. That pressure for a given temperature at which the vapor and the liquid can exist in stable equilibrium.

Secondary air. Air for combustion introduced over the firebed to oxidize the volatile constituents of the fuel as distinguished from primary air introduced below the firebed.

Sensible heat. That heat which when added or subtracted results in a change of temperature, as distinguished from Latent heat, which see.

Smoke. Particles of carbon resulting from incomplete combustion of fuels or other materials containing carbon, and less than one tenth micron in diameter.

Specific enthalpy. See Enthalpy.

Specific volume. The volume occupied by a unit of air. Usually given in cubic feet per pound.

Self-contained cooling unit. A combination of apparatus for room cooling complete in one package; usually consists of compressor, evaporator, condenser, fan, motor and air filter. Requires connection to electric line.

Specific heat. The heat absorbed by a unit of weight of substance per unit temperature rise.

Specific humidity. Weight of water vapor per pound of dry air.

Silicosis. A disease of the lungs arising from the presence of foreign matter in the air, particularly prevalent where silicon is present in the air.

Specific speed. The r.p.m. at which a fan, pump, or turbine would run if it were of such diameter as to deliver a unit volume of fluid per minute against a unit pressure.

Split system. Historically, a combination of warm air heating and radiator heating; the term is also used for other combinations such as hot water - steam, steam - warm air, etc.

Square foot of radiator. The amount of heating surface in the form of radiators, convectors, unit heaters or other devices which will emit 240 Btu per hour. This amount of surface has no exact relationship to the actual area of the surface.

Squirrel cage fan. A centrifugal blower with forward-curved blades.

Stack effect. The impulse of a heated gas to rise in a vertical passage, as in a chimney, a small enclosure, or a building.

Stack height. The height of a convector enclosure measured from the bottom of the enclosure to the top of the outlet.

Standard air. Air weighing 0.07488 lb. per cu. ft.

Stat. A suffix meaning apparatus to render stationary. For example, thermostat, a device to maintain a constant temperature; pressurestat, device to maintain constant pressure, etc.

Static pressure. The pressure exerted by a fluid in all directions; the pressure which would tend to burst the container; the pressure exerted by the fluid if stationary.

Steam. Water in gaseous or vapor form.

Steam jet refrigeration. A method of cooling involving the use of steam nozzles to reduce the pressure in a water chamber so that the water boils at a low temperature; heat is thus drawn from the water, which is thus cooled.

Steam trap. An apparatus for allowing water or air to pass but preventing passage of steam.

Subcooling. Cooling of a liquid refrigerant below the condensing temperature.

Suction pressure. Pressure in the suction line or evaporator of a refrigerating system.

Summer degree hour. The product of one hour in time (1) and the difference between the outside dry bulb temperature at that hour when the temperature is over a base temperature. Some correlation has been found between the number of degree-hours over 85F and summer cooling load in residences.

Sun effect. The quantity of heat from the sun tending to heat an enclosed space.

Superheated steam. Steam at a temperature higher than the boiling temperature corresponding to the pressure at which it exists.

Surface conductance. The heat transmitted from (or to) a surface to (or from) the fluid in contact with the surface in a unit of time per unit of surface area per degree temperature difference between the surface and fluid.

Therm. One hundred thousand Btu.

Thermal resistance. Same as Resistance.

Thermal resistivity. Same as Resistivity.

Throw. Distance from air supply opening, measured in the direction of air flow, from the opening to the point where the air velocity is 50 ft. per min.

Ton of refrigeration. The extraction of 200 Btu per hr., 12,000 Btu per hour or 288,000 Btu per day of 24 hr. The last is also called a ton-day of refrigeration.

Total heat. The sum of sensible heat and latent heat in a substance or fluid above a base point. Frequently 32F or 0F.

Total pressure. The sum of the static pressure and velocity pressure of a fluid.

Tubeaxial fan. An airfoil (propeller) or disc-fan within a cylinder and including driving mechanism supports either for belt drive or direct connection.

Turning vane. Curved strips of short radii placed in a sharp bend or elbow in a rectangular duct to direct the air around the bend in streamlined flow.

Unit heater. Of two types: (1) an assembly of encased heating surface with fan and motor and for connection to a source of steam or hot water; (2) an assembly of the above plus a fuel burner so that the device is for connection to a source of oil or gas (or supplied with coal) and not to steam or water lines.

Two-pipe steam system. A heating system in which one pipe is used for supplying steam to radiator and another pipe is used to return cooled water or condensate to boiler.

Unit ventilator. A unit heater (which see) of type (1) but connected to a source of ventilation air and usually provided with an air filter.

Vacuum system. A two-pipe steam heating system equipped with vacuum pumps to permit maintenance of pressures below atmospheric within the radiators.

Vaneaxial fan. An airfoil (propeller) or disc fan within a cylinder and equipped with air guide vanes either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.

Vapor. The gaseous state of a substance.

Vapor barrier. A material intended to prevent the passage of water vapor through a building wall so as to prevent condensation within the wall.

Vapor pressure. The pressure of the vapor of the liquid at any given temperature at which the vapor and liquid phases of the substance can exist in equilibrium.

Vapor system. A steam heating system operating at pressures very near that of the atmosphere.

Velocity pressure. The pressure exerted by a moving fluid in the direction of its motion. It is the difference between the total pressure and the static pressure.

Ventilation. The art or process of supplying outside (so-called fresh) air to or removing air from an enclosure.

Venturi. A contraction in a pipe line or duct to accelerate the fluid and lower its static pressure. Used for metering and other purposes.

Viscosity. The property of a fluid by which it resists an instantaneous change of shape or arrangement of its molecules. Broadly, resistance to flow.

Warm air furnace. A vessel of steel or cast iron in which fuel is burned and over the surfaces of which air is passed and thus heated. In the heating trade designated as a furnace and not to be confused with a boiler.

Warm air heating system. One whereby air is heated directly in a furnace and circulated to rooms to be heated by gravity or by a fan. The latter is termed Forced warm air heating.

Warming-up allowance. An addition to the capacity of heating system (as calculated for heat loss) to provide for quick pick-up in the morning.

Water line. The height of the normal water level in a boiler measured from the floor.

Wet-bulb temperature. The temperature of the air as measured by a wet-bulb thermometer, and which is lower than the dry-bulb temperature (for all cases except when the air is saturated) in inverse proportion to the humidity.

Wrought iron. A low carbon iron containing a relatively high proportion of residual slag which gives it ductility and toughness.

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DEGREE DAYS (65F BASE) FOR RECENT HEATING SEASONS

Degree day data by months for the heating seasons from 1927 through 1936 for a large number of cities appear in the Degree-Day Handbook and for the heating seasons from 1937 through 1941 on Reference Data sheets 209-210 and 211-212. On this Reference Data sheet and on Reference Data sheet 329-330, are presented similar data for the same cities for the heating seasons from 1941 through 1946.

Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
ALBANY, N. Y.											
1941-42	128	435	683	1097	1332	1237	836	464	162	43	6417
1942-43	172	420	799	1313	1463	1144	1039	759	234	20	7363
1943-44	190	466	871	1297	1173	1203	1116	694	112	52	7174
1944-45	127	434	786	1277	1584	1124	664	437	360	118	6911
1945-46	111	502	759	1354	1265	1206	661	608	312	108	6886
ATLANTA, GA.											
1941-42	—	44	373	531	706	712	410	119	25	—	2920
1942-43	37	81	335	670	586	510	470	176	10	—	2875
1943-44	23	170	411	644	635	446	395	181	34	—	2939
1944-45	4	134	422	754	683	473	162	108	106	1	2847
1945-46	1	132	361	829	657	491	183	102	22	—	2778
BALTIMORE, MD.											
1941-42	8	134	393	698	905	855	554	234	21	4	3806
1942-43	50	149	484	928	932	732	634	437	63	1	4410
1943-44	54	273	536	863	831	783	719	361	25	6	4551
1944-45	27	246	515	936	1072	735	339	216	128	28	4242
1945-46	19	206	464	980	856	751	377	290	73	18	4034
BIRMINGHAM, ALA.											
1941-42	—	24	355	472	689	611	332	89	19	—	2591
1942-43	35	70	269	568	552	444	415	124	1	—	2478
1943-44	20	173	487	679	631	406	309	168	43	—	2916
1944-45	—	89	390	681	683	442	162	111	74	—	2632
1945-46	1	140	340	798	618	469	186	83	13	4	2652
BOSTON, MASS.											
1941-42	89	299	511	923	1132	1058	749	463	164	61	5449
1942-43	103	288	646	1132	1226	967	902	649	243	18	6174
1943-44	120	355	648	1097	1051	1030	956	608	121	76	6062
1944-45	90	363	657	1062	1280	965	585	388	313	94	5797
1945-46	71	368	584	1133	1133	1050	561	565	237	74	5776
BUFFALO, N. Y.											
1941-42	73	382	640	966	1264	1234	893	551	283	56	6342
1942-43	154	367	739	1233	1357	1096	1076	842	411	69	7344
1943-44	191	512	827	1188	1095	1156	1084	721	130	58	6962
1944-45	119	467	695	1228	1514	1088	641	476	447	145	6820
1945-46	125	470	727	1277	1166	1130	633	620	312	128	6588
CHICAGO, ILL.											
1941-42	51	266	653	889	1222	1076	787	370	233	47	5594
1942-43	193	369	723	1307	1312	1042	1019	583	301	45	6894
1943-44	167	363	863	1163	1041	1044	1014	590	136	43	6424
1944-45	62	304	610	1227	1327	980	470	461	331	112	5884
1945-46	103	362	710	1254	1137	981	562	416	279	95	5899
CINCINNATI, OHIO											
1941-42	24	170	561	794	1043	967	599	279	118	13	4568
1942-43	135	255	548	1046	951	824	783	458	102	5	5107
1943-44	88	289	679	1022	894	786	764	347	58	10	4937
1944-45	41	291	611	1110	1204	858	347	274	196	52	4984
1945-46	24	292	578	1131	910	738	299	275	99	13	4359
CLEVELAND, OHIO											
1941-42	53	307	598	826	1135	1109	775	389	182	25	5399
1942-43	135	296	635	1134	1153	964	883	649	240	24	6113
1943-44	134	401	737	1081	987	983	959	596	108	24	6010
1944-45	90	374	618	1167	1398	989	533	400	360	106	6035
1945-46	75	389	626	1190	1109	956	503	508	247	76	5679
COLUMBUS, OHIO											
1941-42	26	214	594	833	1103	1018	661	325	138	17	4929
1942-43	133	246	607	1109	1026	897	833	540	126	8	5525
1943-44	111	339	684	1066	962	864	854	428	49	10	5367
1944-45	53	315	637	1155	1283	903	432	332	241	56	5407
1945-46	31	347	623	1192	1016	862	390	390	169	28	5048
DENVER, COLO.											
1941-42	166	461	620	932	1087	1138	833	399	297	65	5998
1942-43	153	423	657	867	938	682	884	291	387	80	5362
1943-44	93	344	680	927	1011	921	941	651	226	48	5842
1944-45	87	306	688	1001	968	847	676	683	253	142	5651
1945-46	219	315	721	1006	954	764	575	265	405	67	5291
DES MOINES, IA.											
1941-42	78	287	692	986	1285	1128	796	282	203	30	5767
1942-43	193	318	738	1314	1428	998	1022	458	249	36	6754
1943-44	160	374	880	1165	1038	1048	1034	568	139	34	6440
1944-45	70	284	698	1344	1357	1052	552	451	291	87	6186
1945-46	124	350	770	1414	1262	939	510	282	224	53	5928
DETROIT, MICH.											
1941-42	61	329	661	912	1213	1186	820	401	226	35	5844
1942-43	188	365	717	1223	1342	1034	967	679	275	36	6826
1943-44	171	447	804	1165	1062	1053	1050	665	144	28	6589
1944-45	91	388	676	1250	1465	1046	534	448	397	116	6411
1945-46	109	433	699	1250	1141	1077	589	517	273	88	6176

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Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
DULUTH, MINN.											
1941-42	261	588	1001	1319	1489	1369	1041	619	542	259	8488
1942-43	357	580	1084	1633	1842	1409	1454	806	464	243	9872
1943-44	374	565	1179	1455	1335	1468	1341	841	457	237	9252
1944-45	279	556	924	1545	1728	1407	940	864	606	315	9164
1945-46	344	654	1137	1652	1658	1536	906	668	548	233	9336
EL PASO, TEX.											
1941-42	—	72	349	524	537	473	337	97	1	—	2390
1942-43	7	69	216	537	639	389	289	54	5	—	2205
1943-44	14	61	412	664	748	452	339	112	10	—	2812
1944-45	—	49	412	629	594	379	334	148	1	—	2546
1945-46	28	84	310	607	742	462	287	13	5	—	2538
ERIE, PA.											
1941-42	59	297	587	866	1168	1159	854	407	215	39	5651
1942-43	120	315	665	1138	1217	1028	931	725	303	42	6484
1943-44	137	432	757	1106	1044	1051	1019	671	134	37	6388
1944-45	—	390	603	1124	1413	1022	574	419	399	129	6073
1945-46	83	381	630	1162	1066	1026	572	549	288	98	5855
EVANSVILLE, IND.											
1941-42	19	154	598	779	1069	886	559	247	106	9	4426
1942-43	126	246	509	995	964	756	773	357	51	3	4780
1943-44	84	284	668	1013	889	750	688	317	83	12	4788
1944-45	21	273	558	1084	1096	811	339	249	190	36	4657
1945-46	25	277	569	1113	960	725	275	218	122	21	4305
FORT WAYNE, IND.											
1941-42	58	319	708	947	1225	1181	810	379	200	34	5861
1942-43	184	373	700	1294	1295	1027	976	634	239	21	6743
1943-44	203	436	856	1197	1078	1047	1021	609	128	31	6606
1944-45	101	419	724	1334	1456	1034	531	450	390	84	6523
1945-46	102	464	741	1314	1197	1022	524	481	275	71	6191
GRAND RAPIDS, MICH.											
1941-42	70	338	672	909	1216	1180	839	374	215	30	5843
1942-43	192	385	720	1195	1348	1053	1034	655	291	44	6917
1943-44	188	438	825	1155	1053	1057	1057	625	175	38	6611
1944-45	100	418	693	1219	1390	1039	523	470	382	105	6339
1945-46	150	462	741	1229	1175	1102	587	487	278	82	6293
HARTFORD, CONN.											
1941-42	104	343	596	986	1237	1088	755	433	120	34	5696
1942-43	110	334	703	1200	1256	1037	935	665	194	22	6456
1943-44	129	405	752	1183	1106	1063	962	598	86	43	6327
1944-45	87	430	741	1186	1390	1042	589	357	292	95	6209
1945-46	77	432	678	1270	1196	1105	599	549	262	74	6242
INDIANAPOLIS, IND.											
1941-42	27	187	581	822	1105	1016	651	275	131	15	4810
1942-43	133	266	582	1140	1141	932	906	518	158	8	5784
1943-44	112	306	745	1066	942	877	830	423	82	11	5394
1944-45	34	292	625	1178	1255	884	390	337	235	52	5282
1945-46	47	323	645	1205	1050	857	366	333	161	28	5015
KANSAS CITY, MO.											
1941-42	43	182	561	778	1038	923	624	185	134	22	4490
1942-43	136	218	553	1059	1134	738	819	254	153	2	5066
1943-44	60	283	698	1069	898	847	801	448	114	9	5227
1944-45	23	145	556	1142	1061	840	423	326	211	52	4779
1945-46	62	253	566	1216	995	651	300	183	136	27	4389
LOS ANGELES, CALIF.											
1941-42	4	71	74	254	208	259	192	168	98	13	1341
1942-43	3	36	106	209	229	174	194	130	44	42	1167
1943-44	—	49	52	255	261	329	179	184	98	64	1471
1944-45	5	34	182	164	259	244	336	178	110	40	1552
1945-46	2	27	138	246	224	288	248	118	126	—	1417
LOUISVILLE, KY.											
1941-42	7	129	522	713	1000	891	515	214	103	6	4100
1942-43	107	208	483	960	896	726	707	371	47	5	4510
1943-44	61	234	618	928	824	687	682	283	59	3	4379
1944-45	16	229	537	1023	1070	775	291	230	171	39	4381
1945-46	12	238	513	1031	899	670	252	222	89	16	3942
MEMPHIS, TENN.											
1941-42	2	77	484	615	873	699	364	113	33	1	3261
1942-43	61	99	311	704	732	522	567	161	3	—	3160
1943-44	21	172	495	755	710	463	409	179	48	—	3252
1944-45	—	140	404	845	797	584	213	135	94	3	3215
1945-46	11	161	386	862	744	507	194	79	29	4	2977
MILWAUKEE, WIS.											
1941-42	105	388	751	1011	1400	1200	885	469	330	105	6644
1942-43	238	456	780	1391	1439	1178	1116	671	380	104	7753
1943-44	235	456	930	1238	1156	1145	1100	733	261	64	7318
1944-45	82	430	709	1403	1476	1126	643	562	461	194	7086
1945-46	170	440	793	1349	1307	1121	703	540	385	149	6957

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DEGREE DAYS (65F BASE) FOR RECENT HEATING SEASONS

Degree day data by months for the heating seasons from 1927 through 1936 for a large number of cities appear in the Degree-Day Handbook and for the heating seasons from 1937 through 1941 on Reference Data sheets 209-210 and 211-212. On this Reference Data sheet and on Reference Data sheet 327-328 are presented similar data for the same cities for the heating seasons from 1941 through 1946.

Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
MINNEAPOLIS, MINN.											
1941-42	137	440	854	1203	1379	1241	903	400	290	56	6903
1942-43	282	457	944	1535	1771	1333	1286	621	321	69	8619
1943-44	236	480	1066	1287	1194	1290	1212	659	193	42	7659
1944-45	134	431	819	1417	1621	1326	784	634	403	158	7727
1945-46	227	506	997	1602	1544	1352	696	422	320	84	7750
NASHVILLE, TENN.											
1941-42	1	75	490	608	889	765	429	147	50	—	3454
1942-43	73	140	387	791	760	609	610	252	18	—	3640
1943-44	36	213	548	816	772	537	503	241	54	1	3721
1944-45	1	208	494	896	900	649	234	166	130	21	3699
1945-46	7	217	455	953	793	587	198	132	56	12	3410
NEW HAVEN, CONN.											
1941-42	72	272	518	898	1131	1035	734	431	139	32	5262
1942-43	91	285	625	1130	1166	952	859	647	193	26	5974
1943-44	111	386	712	1111	1061	1042	949	601	138	45	6156
1944-45	75	391	694	1101	1294	994	653	389	313	92	5996
1945-46	64	387	640	1182	1122	1033	663	567	283	96	6037
NEW ORLEANS, LA.											
1941-42	—	5	173	211	420	348	166	26	1	—	1350
1942-43	5	17	101	255	361	235	210	21	—	—	1205
1943-44	—	60	199	349	369	115	100	40	2	—	1234
1944-45	—	13	114	314	334	158	22	30	2	—	987
1945-46	—	12	94	404	336	195	64	3	—	—	1108
NEW YORK, N. Y.											
1941-42	26	208	464	834	1056	993	680	360	82	24	4727
1942-43	53	202	555	1034	1058	864	791	589	162	11	5319
1943-44	65	317	607	1008	957	936	872	509	93	27	5391
1944-45	35	283	571	1010	1228	857	453	311	204	76	5028
1945-46	33	276	530	1057	963	940	475	457	154	48	4933
NORFOLK, VA.											
1941-42	—	58	285	547	742	732	433	208	5	—	3010
1942-43	24	65	307	701	675	572	505	309	32	—	3190
1943-44	27	170	402	691	723	628	532	247	6	—	3426
1944-45	5	131	420	779	843	586	248	149	73	12	3246
1945-46	—	107	338	819	681	560	280	230	48	5	3068
OKLAHOMA CITY, OKLA.											
1941-42	14	141	446	643	839	679	437	136	67	3	3405
1942-43	66	142	397	753	859	517	627	93	105	—	3559
1943-44	1	145	441	873	780	592	551	229	55	—	3667
1944-45	10	75	404	851	790	673	300	243	97	1	3444
1945-46	45	130	355	903	798	483	280	81	58	14	3147
OMAHA, NEB.											
1941-42	77	313	743	995	1233	1132	813	271	217	20	5814
1942-43	184	310	727	1264	1396	909	985	349	240	41	6405
1943-44	108	381	877	1150	1035	1068	1051	568	148	35	6421
1944-45	62	258	738	1289	1243	1016	588	454	266	89	6003
1945-46	116	315	755	1416	1191	877	462	242	198	47	5619
PEORIA, ILL.											
1941-42	61	243	655	886	1224	1072	721	304	174	34	5374
1942-43	189	318	707	1255	1262	967	923	488	228	25	6362
1943-44	151	354	860	1204	1056	1031	952	529	131	20	6288
1944-45	69	338	661	1328	1398	991	484	404	312	77	6062
1945-46	102	418	766	1346	1170	957	460	343	231	46	5839
PHILADELPHIA, PA.											
1941-42	12	162	439	779	1008	933	628	292	43	13	4309
1942-43	47	198	515	1000	1002	817	699	521	115	7	4921
1943-44	49	279	574	925	903	853	778	440	41	14	4856
1944-45	31	254	530	973	1188	804	388	245	165	53	4631
1945-46	31	245	496	1026	926	829	421	366	101	38	4479
PITTSBURGH, PA.											
1941-42	32	226	568	796	1031	1027	678	310	110	7	4785
1942-43	108	240	601	1054	985	871	776	560	130	6	5331
1943-44	112	394	697	1021	930	894	833	449	38	15	5383
1944-45	64	322	601	1092	1254	867	416	295	235	71	5217
1945-46	62	340	588	1110	956	844	385	412	148	36	4881
PORTLAND, ME.											
1941-42	216	520	752	1103	1368	1211	898	610	328	122	7128
1942-43	202	494	863	1297	1473	1178	1098	804	377	82	7868
1943-44	240	492	828	1366	1325	1273	1113	780	310	174	7901
1944-45	192	542	806	1277	1516	1201	819	542	480	172	7547
1945-46	178	593	819	1328	1401	1285	789	735	415	163	7706
PROVIDENCE, R. I.											
1941-42	76	295	554	945	1117	1030	718	445	124	47	5351
1942-43	84	292	640	1126	1177	942	862	632	203	15	5973
1943-44	112	349	639	1073	1036	1004	911	566	93	49	5832
1944-45	61	331	643	1048	1255	944	538	330	240	81	5481
1945-46	60	352	579	1122	1083	1008	526	525	206	54	5515

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Heating Season	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Total
READING, PA.											
1941-42	35	221	514	842	1083	984	667	304	70	7	4727
1942-43	83	244	598	1070	1044	881	743	556	121	10	5350
1943-44	90	348	650	992	958	916	840	473	38	16	5321
1944-45	50	334	595	1049	1260	880	411	292	197	65	5133
1945-46	44	312	575	1092	993	908	464	399	123	53	4963
RICHMOND, VA.											
1941-42	7	100	413	657	876	786	490	199	7	—	3535
1942-43	49	135	428	837	767	640	548	343	60	—	3807
1943-44	46	229	493	797	787	687	608	270	14	1	3932
1944-45	16	213	483	865	930	648	268	169	110	27	3729
1945-46	3	193	423	945	809	624	320	232	64	4	3617
ROCHESTER, N. Y.											
1941-42	113	409	653	1029	1257	1253	913	460	217	40	6344
1942-43	158	390	767	1260	1379	1081	1025	793	277	40	7170
1943-44	193	507	835	1232	1127	1192	1096	745	135	66	7128
1944-45	134	491	726	1294	1542	1113	660	457	434	140	6991
1945-46	138	467	735	1304	1202	1164	668	618	309	113	6715
ST. LOUIS, MO.											
1941-42	33	141	531	735	1007	868	558	204	91	19	4187
1942-43	116	197	492	982	1022	701	761	312	93	1	4677
1943-44	56	219	637	1006	849	778	735	343	94	4	4721
1944-45	17	173	552	1106	1124	827	356	253	191	42	4641
1945-46	47	223	576	1130	936	682	252	178	112	19	4155
SAN FRANCISCO, CALIF.											
1941-42	79	150	200	360	431	327	305	284	267	206	2609
1942-43	155	134	271	399	405	286	290	282	224	239	2685
1943-44	101	131	172	356	413	386	286	354	255	218	2672
1944-45	149	127	273	337	458	302	409	275	297	147	2774
1945-46	112	121	268	380	424	398	366	293	293	192	2847
SCRANTON, PA.											
1941-42	90	316	610	951	1194	1146	771	385	123	29	5615
1942-43	128	338	705	1190	1212	1025	893	708	194	21	6414
1943-44	157	452	779	1167	1065	1055	981	604	69	34	6363
1944-45	107	467	719	1172	1430	1022	530	384	290	113	6234
1945-46	83	434	662	1196	1093	1050	557	528	212	91	5906
SEATTLE, WASH.											
1941-42	185	322	474	651	688	589	566	366	262	155	4258
1942-43	113	298	581	638	883	553	602	341	322	160	4491
1943-44	81	330	497	681	649	614	589	398	262	149	4250
1944-45	93	231	515	714	637	568	599	485	203	156	4201
1945-46	171	313	559	684	675	576	585	421	166	150	4300
SPOKANE, WASH.											
1941-42	284	543	744	971	1312	951	790	455	380	184	6614
1942-43	120	457	895	1011	1341	889	904	438	393	201	6649
1943-44	92	469	825	1107	1079	942	867	512	262	104	6259
1944-45	152	331	846	1132	1002	837	844	608	250	166	6168
1945-46	266	440	849	1092	1061	832	738	512	244	159	6193
SYRACUSE, N. Y.											
1941-42	139	395	647	1053	1265	1249	859	453	219	49	6328
1942-43	167	399	752	1282	1420	1105	1023	796	261	36	7241
1943-44	209	506	831	1248	1203	1176	1120	715	129	65	7202
1944-45	135	484	734	1279	1568	1083	639	435	391	129	6877
1945-46	134	467	733	1313	1228	1171	656	618	299	123	6742
TOLEDO, OHIO											
1941-42	48	288	666	890	1194	1166	813	369	193	30	5657
1942-43	164	348	683	1224	1285	1023	973	668	251	29	6648
1943-44	194	437	818	1206	1068	1069	1034	654	131	34	6645
1944-45	103	420	719	1315	1496	1051	540	447	394	101	6586
1945-46	92	452	708	1306	1151	1016	573	513	268	76	6155
TRENTON, N. J.											
1941-42	34	215	499	841	1076	977	673	333	61	14	4723
1942-43	66	242	573	1066	1057	872	763	568	136	10	5353
1943-44	88	350	633	1002	970	906	842	482	54	17	5344
1944-45	49	323	597	1028	1240	865	430	276	182	71	5061
1945-46	44	306	540	1079	970	898	488	423	127	48	4923
UTICA, N. Y.											
1941-42	148	467	663	1082	1335	1305	919	483	195	50	6647
1942-43	176	421	821	1310	1510	1061	1079	810	283	18	7489
1943-44	225	512	895	1354	1267	1260	1195	726	140	73	7647
1944-45	151	554	823	1358	1606	1143	717	436	382	131	7301
1945-46	131	548	778	1398	1233	1132	1390	625	326	142	7703
WASHINGTON, D. C.											
1941-42	9	144	438	715	936	861	545	229	23	1	3901
1942-43	59	153	504	937	894	732	610	414	73	1	4377
1943-44	64	278	551	885	855	778	688	342	15	4	4460
1944-45	24	280	526	947	1062	743	329	197	149	30	4287
1945-46	16	240	485	989	863	716	357	285	77	11	4039

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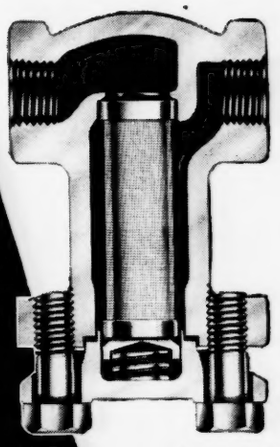
For Ammonia

AND ALL OTHER MATERIALS NON-CORROSIVE TO FERROUS ALLOYS

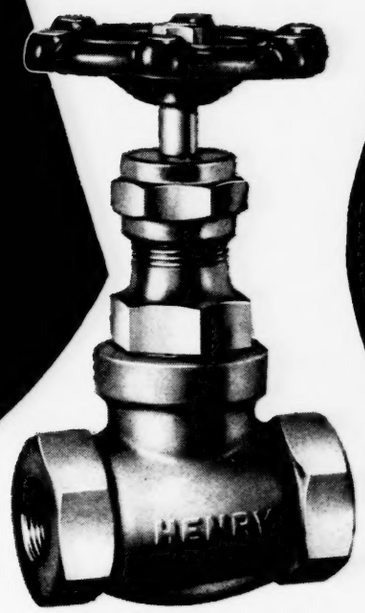
HENRY VALVE CO TRADE MARK

For more than a quarter century, the continued preference shown in the refrigeration and industrial fields for premium-grade Henry Valves, Fittings, and Accessories can be traced to the exclusive patented features of design and construction found in many of the Henry Products.

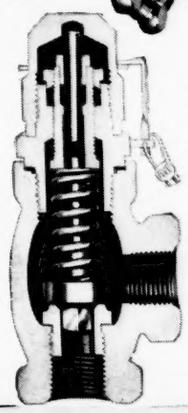


SCALE TRAP OR STRAINER
Large capacity with a spring tensioned self-centering and removable screen. Choice of 50 or 100 mesh. Sizes ¼" to 2" inclusive.

GAUGE GLASS ASSEMBLY
For compressor crank cases and liquid storage tanks. Slotted gauge glass guards. Ball check valve in inlets automatically shuts off flow if the glass should break.



Screwed Bonnet Type Valve



PRESSURE RELIEF VALVE
Accurate release and positive reseating action. Soft alloy metal seat and extended stem permit emergency reseating without removing valve from line. Sizes ⅜" to 2" inclusive.

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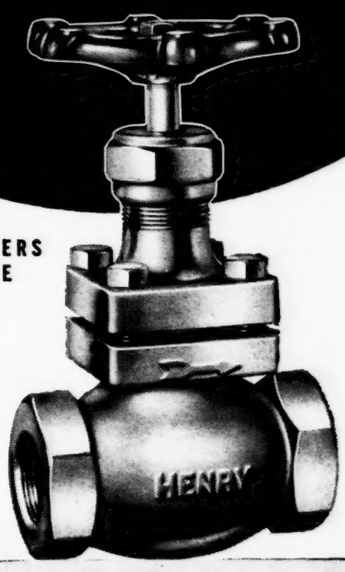

Bolted Bonnet Type Valve

LINE SHUT-OFF VALVES
They are more compact because the bodies are made from die cast permanent mold iron of close grain structure. Two piece stem construction. Floating disc has soft metal alloy insert which rotates on stem. Sizes 1¼" to 2" inclusive.

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PACKLESS AND PACKED VALVES • STRAINERS • DRIERS FOR REFRIGERATION AND AIR CONDITIONING
AMMONIA VALVES • FORGED STEEL VALVES AND FITTINGS FOR OIL, STEAM AND OTHER FLUIDS

Negligence of Operators Responsible For 44% of Boiler Accidents

A RECENT study of boiler and pressure vessel accidents involving loss payments of more than $1,000 each, shows that in a two month period more than 44% of the accidents were the direct result of negligence or incompetence on the part of operators, reports the Hartford Steam Boiler Inspection Insurance Company in a recent issue of its house organ. In all, 36 claims were studied, for which loss payments totaled slightly over $100,000. Of these, 16 accidents were caused by negligence or incompetence, and they resulted in loss payments exceeding $40,000.

The following paragraphs describe briefly the circumstances of the 16 accidents that could have been avoided had operators been a little more aware of the responsibility that was entrusted to them. In noting the approximate loss payments it must be realized that very few of the assureds carried use and occupancy coverage, and that if the values of loss production and lost profits were included they would greatly increase, for most of the cases, the figures that are shown.

Scotch Marine Boiler, Oil Fired

Occurrence: Collapse of furnace
Cause: Low water
Loss: $1,100

Comments: This accident was a direct result of the operator allowing the water to get so low in the boiler that the furnace collapsed.

Water Tube Boiler, Coal Fired

Occurrence: Serious overheating of boiler, cracking of front headers and burning of tubes
Cause: Low water
Loss: $2,000

Comments: This loss was due to incompetence on the part of the boiler operator who allowed the water to get so low in the boiler that serious overheating reuslted.

Water Tube Boiler, Coal Fired

Occurrence: Overheating of boiler damaging approximately 75 tubes
Cause: Low water
Loss: $2,200

Comments: The stoker on this boiler required some repairs and while the fireman was making the repairs the water became low in the boiler. On starting up the stoker the fireman thought he had a full glass of water rather than an empty glass, and before the true condition was discovered and corrected, one tube failed and pulled out of the upper drum. Approximately 75 other tubes were overheated to such an extent that replacement was necessary.

Water Tube Boiler, Coal Fired

Occurrence: Leakage developed at practically all the tube ends
Cause: Low water
Loss: $1,300

Comments: The operator had not checked the water level in this boiler for some time prior to the discovery that a number of tubes were leaking. The boiler was equipped with high and low water alarms, and the operator admitted that he heard one of the alarms, on the boiler blowing but did not pay any attention to it as he believed that it was the high water alarm which was operating.

Horizontal Tubular Boiler, Coal Fired

Occurrence: The boiler overheated to the extent that the shell and heads were badly distorted, some tubes in the bottom rows were pulled out of the rear tube sheet and all other tubes were loosened.
Cause: Low water
Loss: $1,600

Comments: The operator, who was intoxicated, opened up the bottom blow-off valve and drained the boiler completely while it was in service.

Horizontal Tubular Boiler, Coal Fired

Occurrence: The bottom of the shell bulged to such an extent that it was cheaper to replace the boiler than to repair it
Cause: Overheating as the result of oil and grease entering the boiler
Loss: $2,700

Comments: For some weeks prior to the occurrence, the operator had noticed oil in the water glass but had made no attempt to determine its source or the quantity of oil that was entering the boiler. It was later discovered that leakage had developed in a jacketed cooker, and as a result of this, oil and tallow became mixed with the returning condensate.

Water Tube Boiler, Coal Fired

Occurrence: Overheating, bulging and rupturing of water wall tubes
Cause: Scale accumulation
Loss: $2,000

Comments: This accident was due to incompetence or inexperience on the part of the boiler operators who lost control of the feed water treatment, allowing an accumulation of scale to build up in the water wall tubes. The feed water treatment at this plant was under the supervision of a competent feed water consultant, but the plant personnel had not followed his instructions.

Water Tube Boiler, Oil Fired

Occurrence: Tubes burned
Cause: Low water
Loss: $1,100

Comments: The water tender, whose only duty was to maintain the water level in the boiler, neglected to watch it and a low water condition developed.

Horizontal Tubular Boiler, Coal Fired

Occurrence: Serious overheating and burning of boiler
Cause: Low water
Loss: $5,000

Comments: This boiler was tended during the night by a fireman who also acted as a watchman, making rounds of the plant at hourly intervals. At about the time for him to make one of his rounds, a sprinkler head over the boiler let go. He thought that the boiler had been flooded and the safety valve was popping. He cut off the feed water, opened the blow-off valve, and made his rounds as a watchman. Upon his return to the boiler room, water was still running down over the boiler, and he then noted that the sprinkler head had failed. During this time, the boiler had been so seriously overheated and burned that it had to be practically rebuilt.

Locomotive Type Boiler, Gas Fired

Occurrence: Crown sheet collapsed
Cause: Low water
Loss: $6,500

Comments: This occurrence was directly caused by negligence on the part of the operator who failed to maintain water at the proper level in the boiler, resulting in serious overheating and failure.

Water Tube Boiler, Coal Fired

Occurrence: A number of tubes seriously overheated and one tube ruptured violently
Cause: Carelessness on the part of the operator
Loss: $2,500

Comments: A review of the charts from the boiler recording-instruments disclosed that there had been a gradual increase in flue gas and steam temperature during a period of about 25 minutes prior to the tube failure. The rise in temperature was a definite indication that there was a deficiency in the amount of water being fed to the boiler.

Water Tube Boiler, Coal Fired

Occurrence: One tube pulled out of the front upper drum, several tubes badly overheated, and the tube sheet of the front upper drum distorted
Cause: Carelessness on the part of the operator
Loss: $2,900

Comments: A review of the charts from the recording-instruments disclosed that the operator had permitted the water level in the boiler to become dangerously low. It is interesting to note that the water tender had been employed in the assured's power plant more than 20 years and had acted as water tender for the past three years. The fireman had been acting in such a capacity at the plant for more than 12 years. This case is one of carelessness by experienced operators.

Horizontal Tubular Boiler, Wood Fuel

Occurrence: The shell of the boiler bulged, a number of tube ligaments cracked and several tubes were partially pulled from the rear tube sheet
Cause: Carelessness on the part of the operator
Loss: $2,000

Comments: Following the occurrence, the blow-off valve on the boiler was found partially open. It seems evident that the operator neglected to close this valve properly after having blown the boiler down, and as a result the water level became dangerously low.

Sotch Marine Boiler, Gas Fired

Occurrence: Collapse of the furnace
Cause: Carelessness on the part of the operator
Loss: $2,500

Comments: The boiler had been cleaned on the day before the accident. Then, a hydrostatic test had been applied to the boiler, at which time a number of tubes were found to be leaking. These tubes were rolled and after they had been made tight, the water level in the boiler was lowered by opening the blow-off valve. It is believed that the blow-off valve was not properly closed and that the boiler was emptied during the night as a result of leakage through the valve.

Cast Iron Steam Boiler, Coal Fired

Occurrence: Cracking of all sections
Cause: Negligence on the part of the operator
Loss: $1,300

Comments: A leak developed in the system to which the boiler was connected and as a result, sufficient water was voided to create a low water condition. The operator had been aware that a leak had occurred but had ignored its existence.

Cause: Low Water!

McDONNELL